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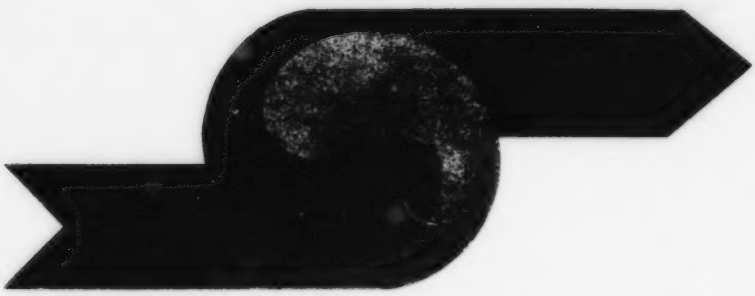
Heat Pump Program
Programme des pompes de chaleur

# *A Restaurant Wastewater Heat Recovery System Utilizing Heat Pump Technology*

E.J. Faraci and Associates

July 1985



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# A RESTAURANT WASTEWATER HEAT RECOVERY SYSTEM UTILIZING HEAT PUMP TECHNOLOGY

A Final Report for Phase B of a Heat Pump Research and Development Project Titled 'The Installation, Monitoring and Evaluation of a Restaurant Wastewater Heat Recovery Utilizing Heat Pump Technology' Sponsored by Energy Division of the National Research Council of Canada.

submitted by:

E.J. Faraci and Associates on behalf of the Keg Restaurants of Manitoba Ltd.

submitted to:

Supply and Service Canada

In accordance with:
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## PREFACE

In February 1982, NRC requested contract proposals for heat pump research and development and the evaluation of conservation potential in order to promote market exploitation of heat pumps in Canada. In the industrial sector, priority was given to proposals which were directed to the specific application of commercially available equipment into areas without previous operating experience. Contracts were awarded for heat pump heat recovery projects in the following industries: pulp and paper, metals mining and smelting, abrasives industry, milk processing, dairy industry, marine food resources, laundry and dry-cleaning, and restaurants.

This report presents the results of the restaurant wastewater heat recovery project. Although, from the perspective of the National Research Council, the project has been concluded, the optimization of the heat recovery system from the perspective of the Keg Restaurant and their consultant continues. Unfortunately the commissioning of the system was not a straight forward experience. The most significant barrier to the optimization of the heat recovery system was the poor performance and unsatifactory operation of the heat pump. Twice, during the monitoring phase, it was necessary to replace the heat pump compressor. Although the third compressor performed satisfactorily, its life was limited to 15 days! Ironically this compressor failed during the last week of March, the termination of the NRC/Keg contract.

In parallel with the preparation of this final report and a conviction that the flavour of the project can and will improve, the consultant again has negotiated the replacement of the compressor with the heat pump supplier.

In retrospect, it is easy for an enthusiastic designer to project an attractive economic scenerio for a 'paper' system. It is difficult, for that same designer to report less than satisfactory performance for the 'real' system. In this report no attempt has been made to smooth over shortcomings in either design or equipment performance in an effort to support unrealistic conclusions.

Although there is much to learn about small capacity, heat pump systems, this project has advanced the Canadian state-of-technology for restaurant wastewater heat recovery. Proof-of-concept has been demonstrated. It is in the interest of the private sector to support the development and commercialization of wastewater heat recovery equipment for the restaurant and foodservice industry.

## EXECUTIVE SUMMARY

In September, 1983, a contract to install, monitor and evaluate the performance and economics of a restaurant wastewater heat recovery system was awarded by Supply and Services Canada on behalf of the National Research Council to the Keg Restaurants of Manitoba Ltd. This contract followed a previous contract awarded to E.J. Faraci and Associated in September, 1982 for the design and economic evaluation of a wastewater heat recovery system for a Keg restaurant in Winnipeg.

The hot water heating bill for the Canadian restaurant foodservice industry totalled $75 million dollars in 1972. Industry wide application of wastewater heat recovery technology has the potential to cut this bill in half. In the 93 Keg Restaurant operations across Canada, this represents an energy cost saving of $300 000 per year.

The Keg Restaurant selected for this demonstration project was a full service, steak and seafood establishment. The annual utility bill approached $30 000, of which $6 700 was associated with domestic hot water usage. A load profile for the consumption of hot water at the Keg was developed and used as a basis for the design of the wastewater heat recovery system.

The wastewater heat recovery system (WHRS) was centered around a 3 ton (10 kW) rotary compressor heat pump. Included with the system, was a passive water-to-water heat exchanger in series with the heat pump evaporator on the wastewater circuit and the heat pump condenser on the domestic water circuit. A UL listed, double wall heat exchanger and evaporator were specified.

Commissioning the heat recovery system was not a 'text book' experience. Equipment failures included sump pump level control, heat recovery system controls, integrating energy meter, the wastewater pump, and three heat pump compressors.

The system operated satisfactorily for only three weeks, just prior to the failure of the third heat pump compressor and the end of the NRC contract. During this period, the performance was monitored and used to project an annual energy cost saving of $1 300 per year for a 'reliable' system. The measured coefficient of performance (COP) was 4.1 for the heat pump and 3.5 for the system. Unfortunately the annual energy cost saving extrapolated for the system fell short of the $3 000 saving predicted by the design study (primarily due to the poor performance of the passive wastewater-to-domestic water heat exchanger). Thus based on

an estimated cost of $8 000 to install a 'packaged' heat recovery system in a new restaurant (the installed cost of the Keg's research oriented system was $35 000), the payback on a restaurateur's investment quickly increased to six years from the anticipated two and one-half years. Furthermore, if the rate of return had been adjusted to reflect the maintenance burden that had been experienced, there would have been no economic rational for this one-off system. Therefore it was recommended that the Keg management not proceed with further installations until such time as wastewater heat recovery equipment is commercially available. The continued operation of the existing system, contingent on compressor replacement, was recommended.

The project demonstrated that heat pump, wastewater heat recovery was technically viable. However, further research and development is required before the technology can be marketed as a 'package' to the Canadian restaurant and foodservice industry. (One of the primary goals of this demonstration project was to stimulate the development of off-the-shelf wastewater heat recovery systems for commercial kitchens).

## RÉSUMÉ ADMINISTRATIF

En septembre 1983, un contrat pour l'installation, le monitorage et l'evaluation de la performance et de la rentabilité d'un système de récupération de la chaleur des eaux usées (SRCEU) des restaurants a été adjugé par Approvisionnements et Services Canada, pour le compte du Conseil national de recherches, aux Keg Restaurants of Manitoba Ltd. Ce contrat faisait suite à un contrat antérieur adjugé à J.E. Faraci and Associates en septembre 1982 pour la conception et l'évaluation financière d'un SRCEU pour un restaurant Keg à Winnipeg.

La facture d'eau chaude pour l'industrie canadienne de la restauration en 1972 s'est élevée à 75 millions de dollars. L'application, dans toute l'industrie, de la technologie de la récupération de la chaleur des eaux usées pourrait réduire cette facture de moitié. Pour les 93 restaurants Keg au Canada, cela représente une économie d'énergie en argent de 300 000 $ par année.

Le restaurant Keg choisi pour ce projet de démonstration se spécialise dans le steak et les fruits de mer et offre un service complet. Les services publics coûtaient près de 30 000 $ par année, dont 6700 $ en eau chaude domestique. Un profil de la consommation d'eau chaude au Keg a été établi et utilisé comme base pour la conception du SRCEU.

L'élément principal du SRCEU est une thermopompe à compresseur rotatif de 3 tonnes (10 kW). Le système comprend aussi un échangeur de chaleur eau-eau passif monté en série entre l'évaporateur de la thermopompe côté eaux usées et le condenseur de la thermopompe côté eau domestique. Un changeur de chaleur à double paroi et un évaporateur, certifiées par UL, ont été spécifiés.

La mise en service du SRCEU n'a pas été une expérience de tout repos. Plusieurs éléments ont fait défaut : la commande du niveau de la pompe d'assèchement, les commandes du système de récupération de la chaleur, le totalisateur d'énergie, la pompe à eaux usées et trois compresseurs dans la thermopompe.

Le système n'a bien fonctionné que pendant trois semaines, juste avant que le troisième compresseur ne tombe en panne à

la fin du contrat du CNRC. Pendant cette période, la performance a été monitorée, et les résulatats ont été utilisés pour projeter une économie d'énergie en argent de 1300 $ par année pour un système "fiable". Le coefficient de performance (CP) mesuré a été de 4,1 pour la thermopompe et de 3,5 pour le système. Malheureusement, l'économie annuelle extrapolée pour le système a été moindre que les 3000 $ prévus par l'étude théorique (surtout à cause du faible rendement de l'échangeur de chaleur eau usée-eau domestique passif). Par conséquent, si on estimait à 8000 $ le coût d'installation d'un ensemble SRCEU dans un nouveau restaurant (le coût installé du système expérimental Keg a été de 35 000 $), la période de recouvrement du coût du restaurateur passerait rapidement de 2,5 ans comme prévu à 6 ans. En outre, si le taux de rendement avait été ajusté pour refléter les frais d'entretien, il n'y aurait eu aucun avantage financier à investir dans un tel système. Il a doc été recommandé à la direction des restaurants Keg de ne pas répéter l'expérience jusqu'à ce qu'un SRCEU soit disponible dans le commerce, mais de continuer d'exploiter le système existant, quitte à remplacer le compresseur au besoin.

Le projet a démontré que la récupération de la chaleur des eaux usées au moyen d'une thermopompe est techniquement réalisable. Cependant, il faut poursuivre la recherche et le développement avant que la technologie puisse être commercialisée sous forme d'un "ensemble" dans l'industrie canadienne de la restauration. (Un des principaux objectifs de ce projet de démonstration était de stimuler la mise au point de SRCEU prêts à installer dans des cuisines commerciales).

## 1.0 INTRODUCTION

It has been estimated[2] that the energy bill for the Canadian restaurant and foodservice industry totalled $500 million in 1982. Detailed energy audits [3],[12] indicate that domestic hot water accounts for 5 to 25% of the energy consumed by foodservice operations. Assuming an average of 15%, this translates to a hot water energy bill of $75 million and an energy consumption of 12 million gigajoules (GJ).

The industry wide application of restaurant wastewater heat recovery, utilizing heat pump technology, could be expected to reduce restaurant hot water energy consumption up to 50%. This translates to a national saving of 6 million gigajoules per year, equivalent to more than 800,000 barrels of oil. This represents more than 1% of the production from the oil sands projects and could be considered significant.

Phase A of this heat pump application study comprised the design and engineering evaluation of a heat pump controlled, wastewater heat recovery system. The design study concluded that a restaurant wastewater heat recovery system utilizing heat pump technology was feasible and potentially cost effective, assuming the commercial development of 'packaged' heat recovery units. However the feasibility study emphized that:

1. the actual performance of energy conserving equipment and systems often falls short of the performance claimed by the manufacturer or the theoretical performance projected by the designer.
2. the "maintenance" associated with a wastewater heat recovery system may not be acceptable to the average restaurant operator.

The literature and product review conducted during the design phase did not identify any commercially available wastewater heat recovery equipment for the foodservice industry. However, during the course of the project, product literature for two dishwasher heat recovery systems was acquired. The first unit (Appendix 3) was an add-on heat exchanger for a commercial dishwasher. The second (Appendix 4) was a Swedish made dishwasher with an integrated heat recovery system. Such commercial developments support the conclusions of the design study.

The objective of Phase B was to install and monitor a 'research' oriented, wastewater heat recovery system at a Keg restaurant in order to establish the practicality of grey water heat recovery in the restaurant industry.

The scope of Phase B included project tendering, system installation and commissioning, performance monitoring, system evaluation and project reporting.

## 2.0 THE KEG

In 1982 'Keg Restaurants' were ranked[11] as the 7th largest enterprise in the Canadian foodservice industry based on the gross food and beverage sale of $112 million from 75 restaurants. By 1984 their operation had expanded to 93 outlets across Canada. The installation of wastewater heat recovery equipment throughout the Keg operation could represent an energy recovery of 30 000 GJ and a potential energy cost saving approaching $300 000 per year.

The Keg restaurant selected for this demonstration project (Photo A1) is a licensed steak and seafood establishment with a seating capacity of 160 in the dining area and 75 in the lounge. The cleaning and food preparatory activities usually commence at 7:30 am. The lounge opens at 4:30 pm and the dining room shortly thereafter. The restaurant operation ends by 2:00 am.

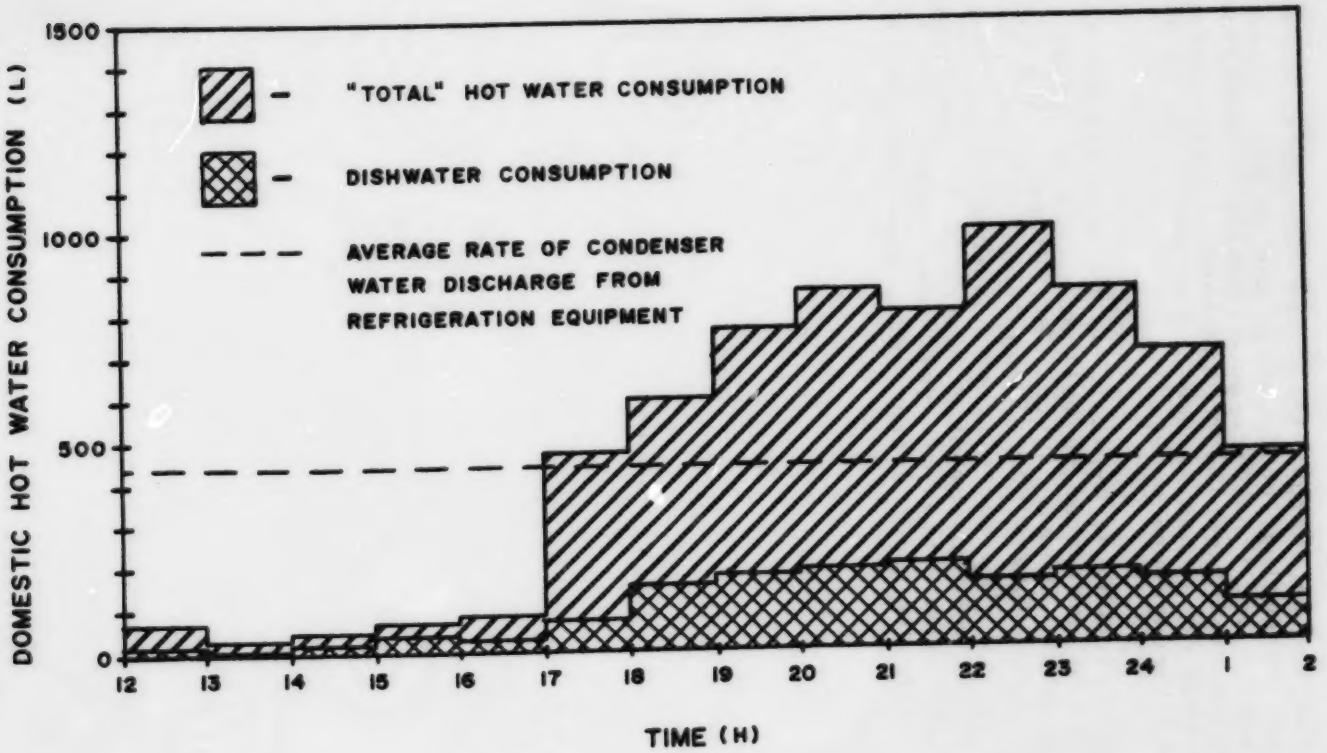
Natural gas provides energy for space heating, hot water heating, makeup air heating, clothes drying and two kitchen ranges. The Keg utility bill for 1984 totaled $29,000; of which $13,400 was electricity, $9,500 was natural gas and $6,100 was a water and sewer charge. The associated energy consumption was 1440 GJ of electricity and 1950 GJ of natural gas at a rate of $9.33/GJ and $4.85/GJ respectively. The combined consumption of energy was 3400 GJ at an average cost of $6.75/GJ.

The Keg hot water consumption was monitored throughout the commissioning period and an average consumption of 7200 litres per day recorded. This was 2800 litres less than the consumption recorded during 1982, primarily due to the replacement of a continuous flow glass washer in the bar with a manual loading type. The associated energy cost saving was approximately $2,000 per year, quickly returning the investment on the new glass washer. Based on the mean water main temperature of 10 C for Winnipeg and a hot water tank set point of 65 C, a total hot water heating load of 550 GJ was calculated. Applying a DHW energy conversion efficiency of 50%, this hot water load translated to a natural gas input of 1100 GJ at a 1984 cost of $5,340. This represented 56% of the natural gas consumed by the Keg operation, or 23% of the total energy cost.

With the installation of a water meter on the supply line to the domestic hot water (DHW) tank and a gas meter on the supply line to the DHW tank, (Photo A2) it was possible to accurately determine the overall energy conversion efficiency (ie. 1 - combustion inefficiency + stand-by losses) of the existing DHW sytem. The efficiency calulated for the Keg DHW system, based on known input energy and water consumption averaged 50%. (Significantly lowerer than the 70 or 75% often assumed by designers when estimating the energy consumption of a DHW system!)

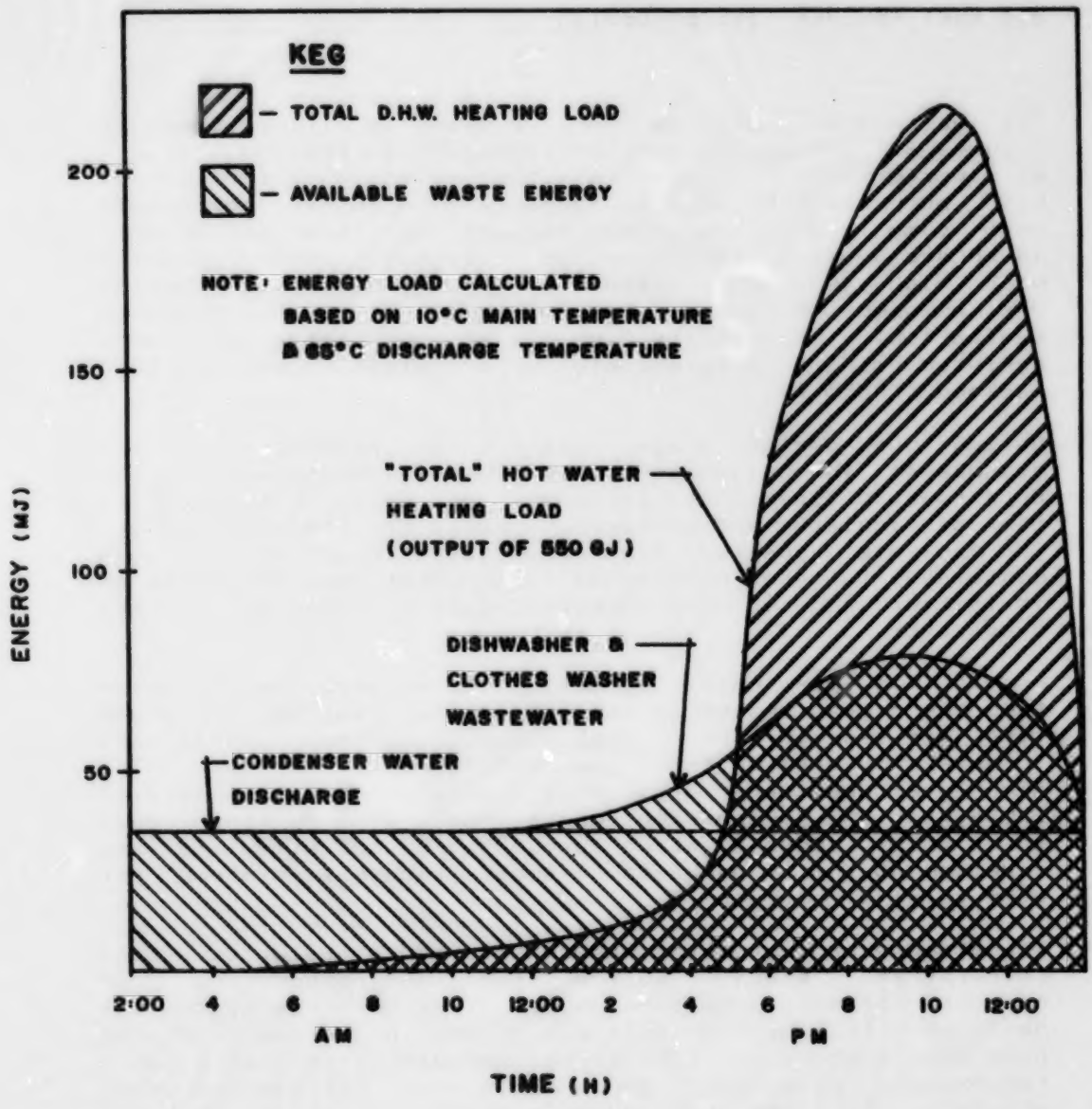
In addition to the energy that was dumped to sewer by the hot water consuming equipment, the walk-in freezer and refrigerator and the ice-cube making machines all incorporated water cooled condensers. Field measurements indicted that more than 10 000 L/d of water was rejected at an average temperature of 32 C.

Figure 2.1 shows a typical daily profile for the total consumption of hot water as well as the hot water used by the dishwasher (Photo A3). As indicated, the dishwasher consumed approximately 20% of the total. Also imposed on the figure is the quantity of condenser water rejected by the refrigeration equipment. Figure 2.2 compares, as a function of the time of day, the energy available for recovery with the input energy requirement of the DHW tank.




AVERAGE DAILY HOT WATER LOAD PROFILES

FIGURE 2.1




SUPPLY & WASTEWATER PROFILES

FIGURE: 2.2

## 3.0 HEAT RECOVERY SYSTEM DESIGN

The wastewater heat recovery system is illustrated in Figure 3.1 and photographs A4 through A10. The manufacturer's performance ratings for the selected 3 ton (10 kW) heat pump are tabulated in Appendix 2. A rotary compressor heat pump was specified because of it's performance advantage over a reciprocating compressor when operating with high (eg. 30°C) evaporator temperatures. The system was designed to accomodate 'on-site' research and system evaluation. Flexiblity in operating strategies were achieved using a programmable direct digital control (DDC) package (Photo A).

During the design stage, waste water samples from the dishwasher and glasswasher had been analyzed by the Department of Chemistry, University of Manitoba to quantify the particulate and grease components of the wastewater. The analysis suggested that the large quantity of soap contained in the dishwasher wastewater was acting as an emulsifier and should suppress grease fouling of the evaporator.

As it was physically impractical to re-pipe the discharge from the glass washer in the lounge, the wastewater sources were limited to the dishwasher, clothes washer and condenser water from the refrigeration units. The dishwasher wastewater source is typical of all full service restaurant and foodservice operations. (The ultimate goal of this demonstration project is to stimulate the development of packaged heat recovery systems for commercial dishwashers.)

The dishwasher wastewater was considered a relatively high temperature energy source from the perspective of conventional heat pump technology. Therefore the system was designed with a passive heat exchanger in series with the heat pump evaporator. (The dishwasher discharges water at a temperature exceeding 65°C, while the performance data presented for the heat pump (Appendix 2) implies that the unit should be operated within a range of evaporation inlet temperatures from 5°C to 50°C.)

A heat pump requires that the refrigerant temperature and pressure in the evaporator are lower than the temperature and pressure in the condenser. If these conditions are not maintained during heat pump operation, the refrigerant will tend to drive the compressor and the motor will overload. As a consequence, the mean temperature of the wastewater that is circulated through the evaporator heat exchanger should not be allowed to exceed the mean temperature of the domestic water in the condenser. With possible temperature variations in the restaurant wastewater from approximately

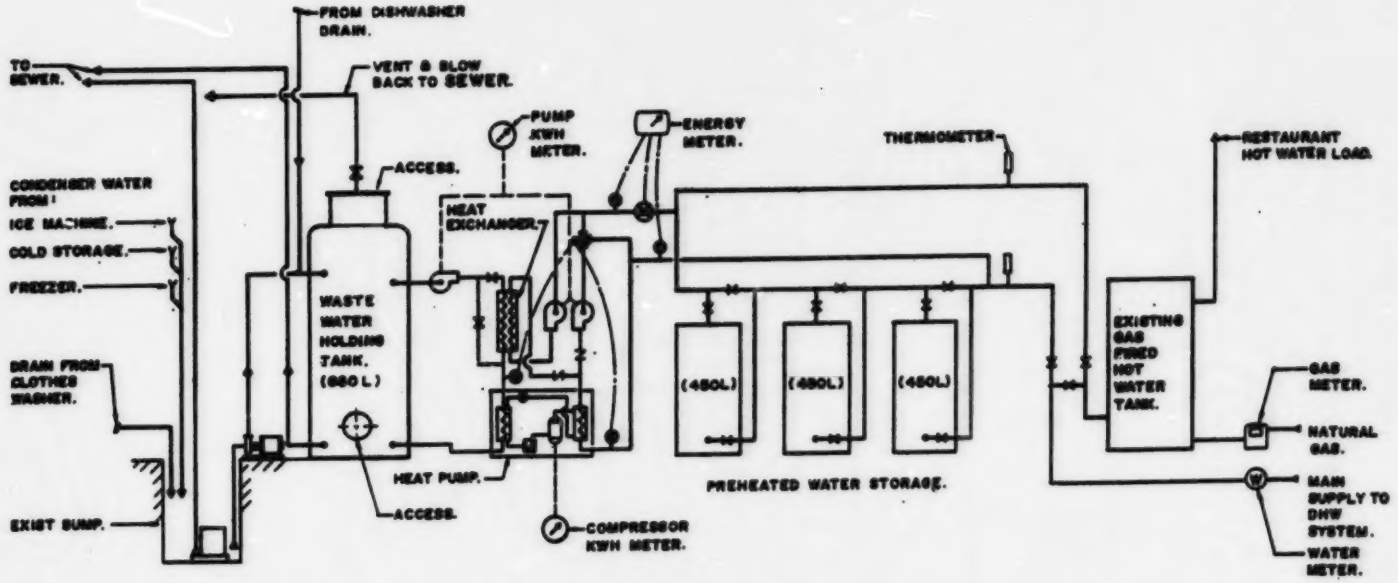
15°C to 70°C and preheat water from 10°C to 50°C, such adverse operating conditions frequently would be encountered. Unfortunately, these wastewater and preheat water temperature fluctuations are not in phase. The system operated with high wastewater and low preheat water temperatures during periods when the restaurant hot water usage was high. This temperature relationship was reversed as the consumption of hot water decreased.

A 3-way valve and recirculation loop on the condenser water circuit was incorporated to ensure that the temperature of the domestic water leaving the condenser was always higher than the temperature of the wastewater entering the evaporator.

The system was designed to recover energy in the following operating modes:

1. heat pump only
2. heat exchanger only
3. heat pump in series with heat exchanger.

No regulatory barriers were found relating to reclamation of energy from restaurant grey water. Though not addressed explicitly, the Canadian Plumbing Code does not prohibit such fixtures. Municipal codes have over-riding jurisdiction however, and should be consulted accordingly. The study by Pemberton[1] used a closed loop heat pump cycle in his Waterloo Ontario home to transfer heat from grey water to potable water. No regulatory problems were noted for this installation. However a review of heat pump domestic water heating packages manufactured in the United States indicated that all units were equipped with UL listed, double wall heat exchangers. Therefore double wall heat exchangers for both the evaporator and the water-to-water heat exchanger were specified by the design. Such precaution should smooth the way for across-the-board acceptance of 'packaged' restaurant wastewater heat recovery systems by the local authorities in Canada.




THE KEG WASTEWATER HEAT RECOVERY SYSTEM

FIGURE 3.1

## 4.0 OPERATING EXPERIENCE

The commissioning phase of the project was burdened with equipment, control and design related problems.

Initially the dishwasher water drain had been re-piped to an existing sump in the basement (used to collect laundry water and condenser cooling water). From there, the mixture of wastewater was pumped to the wastewater holding tank and then on to the sewer. Unintentionally, the drain from the prewash sinks also had been connected to the sump. Within a short period of time, the sump level control failed (due to heat and grease from the kitchen wastewater) and the basement flooded. This occurred twice, before the level control was changed (it had been permanently damaged) and the dishwasher drain diverted directly to the wastewater holding tank. The drain from the prewash sinks was excluded from the WHRS and returned to the main sewer line.

The first heat pump compressor would not operate because of a seized rotor. Eventually this compressor was replaced and performance monitoring commenced. The input/output keyboard for the direct digital control system then failed, along with the integrating energy meter. Eventually, the system was put into operation and the performance of the heat pump was established at 60% of both rated heating capacity and COP. After several service checks (which included recharging the system with refrigerant and setting the amount of superheat to match factory specifications) and negotiation with the supplier, the second compressor was replaced. By this time, less than one month remained in the contract schedule for performance monitoring. The third compressor performed well for the first three weeks of operation and then failed.

Evaporator and heat exchanger fouling on the wastewater circuit was a design concern. Although the WHRS system did not operate for sufficient time to determine the effect of fouling on system performance, the concern was reinforced by the fact that the wastewater pump plugged after several months of operation in the 'heat exchanger only' mode. (Photo A11). Although pump cleaning was accomplished within 30 minutes, it represented a maintenance burden on the system. Also, the build-up of white sludge on the inside surface of the wastewater piping (Photo A12) suggested that a similar condition may be occurring on the heat transfer surfaces of the evaporator and water-to-water heat exchanger.

Interestingly, heat exchanger cleaning using an acid detergent is described as a maintenance procedure for the Swedish dishwasher (see Trouble Shooting Guide, Appendix 4).

## 5.0 SYSTEM PERFORMANCE

Two methods of monitoring the performance of the WHRS were incorporated. The first comprised an integrating energy meter (Photo A) to record the energy delivered to the DHW system by the WHRS. The second method involved measuring the quantity of natural gas (Photo A2) consumed by the DHW boiler with and without the WHRS operating. The calculated reduction in gas consumption per unit volume of hot water used by the restaurant represented the 'real' saving to the Keg.

Although the energy meter was used to measure the output of the WHRS over short duration tests, the second method was used to evaluate the overall performance (ie. effective system COP) of the installation as it accounted for the efficiency of the DHW system.

The following terminology and equations were used to expess the performance of the wastewater heat recovery system (WHRS):

$Q_{HRS}$ = total energy contribution of WHRS

$Q_{HP}$ = energy contribution of heat pump

$Q_{HX}$ = energy contribution of heat exchanger

$\Delta Q_{DHW}$ = reduction in DHW energy consumption due to $Q_{HRS}$, $Q_{HD}$ or $Q_{HX}$.

$E_{HP}$ = electrical energy consumed by heat pump

$E_P$ = electrical energy consumed by domestic and wastewater pumps

$E_{TOT}$ = total system electrical consumption

$= E_{HP} + E_P$

$COP_{HP}$ = heat pump coefficient of performance

$$= \frac{Q_{HP}}{E_{HP}}$$

$COP_{SYS}$ = heat pump system coefficient of performance

$$\frac{Q_{HP}}{E_{HP}+E_P} \qquad \frac{Q_{HP}}{E_{TOT}}$$

$COP_{EFF}$ = effective system coefficient of performance

$$= \frac{\Delta Q_{DHW}}{E_{TOT}}$$

After the second compressor was commissioned, a 'cold start' test was conducted to compare the performance of the heat pump with the performance tabulated in the product literature (Appendix 2). As illustrated in Figure 5.1, the performance was significantly less than the rated performance. This compressor also was replaced under warranty.

Several months later, the third compressor was installed and commissioned. Performance monitoring of the WHRS in the 'heat pump only' mode commenced immediately, as the performance of the WHRS in the 'heat exchanger only' mode had been established prior to compressor replacement. After an eleven day test, the heat exchanger was valved into the system and performance monitoring continued. Three days later the heat pump compressor malfunctioned and the operation of the WHRS was returned to the 'heat exchanger only' mode.

The performance of the WHRS is summarized in Table 5.1. Although not shown in the table, the heating output of the heat pump averaged 28 MJ/h (8 kW), falling short of the rated capacity by approximately 20%. Also not tabulated, was the measured heat pump COP of 4.1 which closely matched the rated COP. The system COP in the 'heat pump only' mode averaged 3.5 representing an effective COP of 5.6 (accounting for the inefficiency of the DHW system).

Economics

Based on the run-out rate and the demand charge for electricity and the run-out rate for natural gas, an incremental energy cost of $9.30/GJ and $4.50/GJ was calculated respectively. If the WHRS was operated in the 'heat pump only' mode, a annual cost saving of $1,200 per year was estimated. This compared to a saving of $500 per year projected for the WHRS operating in the 'heat exchanger only' mode. Unfortunately an annual saving of only $1,340 was estimated for the combination of the heat pump and the heat exchanger, just slightly more than the saving for the heat pump by itself.

The $1 300 saving was significantly lower than the $3 000 saving predicted by the design study, primarily due to the poor performance of the passive heat exchanger (although the reduced capacity of the heat pump was a contributing factor). Had the heat exchanger performed with an effectiveness of 50% (as specified by the design), half of the energy available in the dishwater should have been recovered (see Figure 2.2). This would have represented a saving of at least $1 000, justifying the $2 500 incremental cost to install the heat exchanger in the circuit.

Another problem with the heat exchanger application occurred when it was operated simultaneously with the heat pump. As the flow capacity of the wastewater pump circulating water through the system greatly exceeded the average rate at which 'hot' dishwater was discharged into the holding tank, the higher temperature water was quickly mixed with cooler water being maintained in the tank by the heat pump. This lowered the availability of energy in the dishwater, reducing the operating temperature difference across the heat exchanger and the associated rate of heat transfer between the waste and domestic water.

Obviously the economic scenario for this heat pump application was affected by the lower heat recovery capacity of the system. However, it must be recognized that the Keg installation had been designed as a field 'research' system. Two-thirds of the $35 000 price tag for the Keg project related directly to the programmable direct digital control system, the energy monitoring instrumentation, and the 'Cadillac' approach to flow circuiting and valving. Another major component of the installed cost related to the fact that the system had been retrofitted.

Ultimately, the cost benefit of applying wastewater heat recovery technology to a restaurant operation relates to the installed cost of a fully developed, packaged system. Thus the design study had estimated that an off-the-shelf system could be installed in a new restaurant for less than $8 000. But even with a capital cost of $8 000, the payback on a restaurateur's investment would exceed six years if the performance of the 'packaged' system was only able to match the performance demonstrated by the Keg project.

No attempt was made to include the maintenance costs as part of the economic analysis, as the operating history for a 'reliable' system had not been established. However, the next section implies that system reliability and maintenance is an aspect that will affect significantly the rate of return on a wastewater heat recovery system investment.

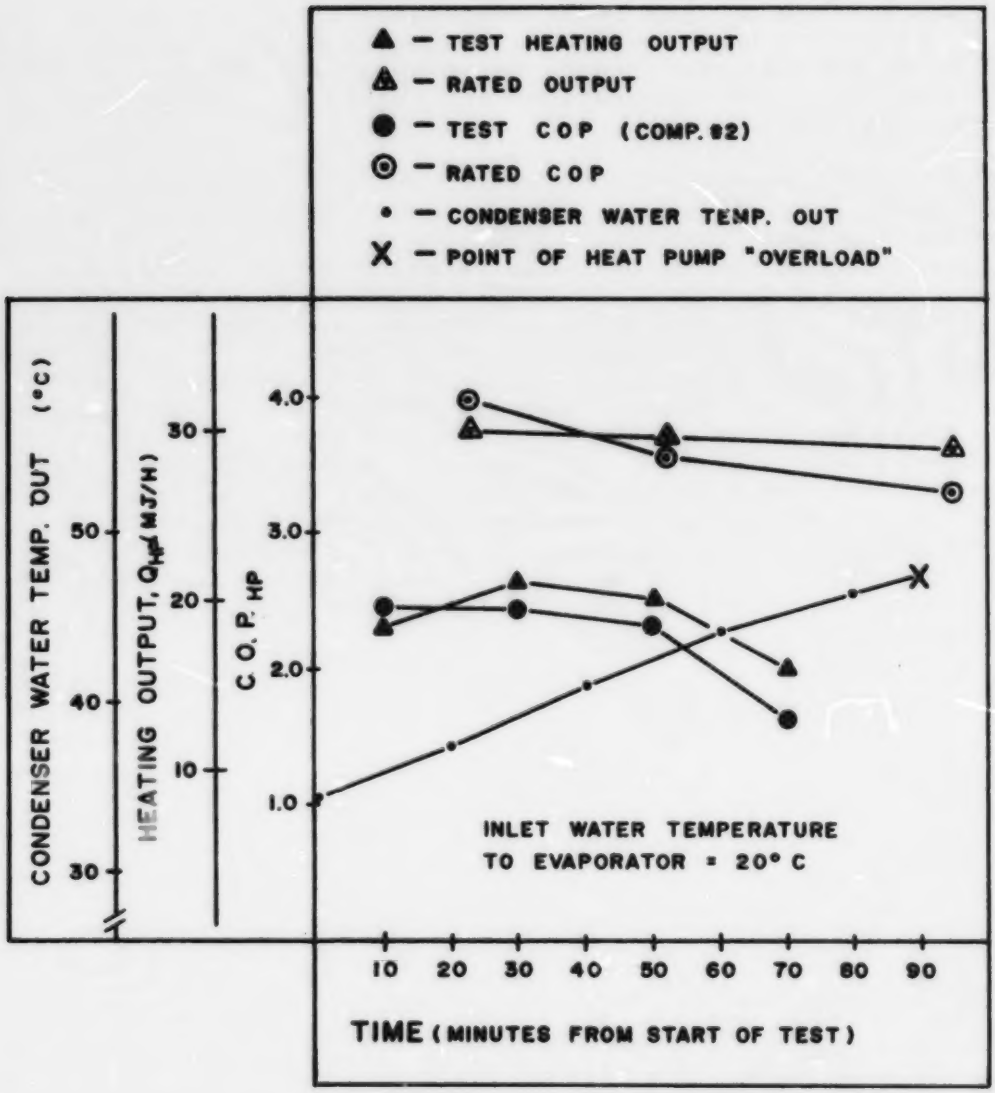
## Heat Pump Reliability

Although it had been acknowledged by the design study that the heat pump reliability and maintenance aspects represented an 'unknown' for the restaurant application, the serious problems that were experienced with compressor operation over the short term had not been anticipated.

The failure of the first compressor to rotate was explained by the fact that the U.S. made heat pump originally had been shipped to Canada for display at a trade show. The long period of inoperation and unknown handling procedures were suspect.

Nor did the second compressor, shipped directly from the Canadian distributor, operate satisfactorily. It was noisy, and as shown in Figure 5.1, performed poorly. The default explanation was simply that it was a defective product.

The failure of the third compressor was more of a concern, however, as it had operated satisfactorily for the first two weeks. Generally, a compressor will fail if it is subjected to highly abusive conditions (eg. extremes of temperature or pressure, loss of oil and/or loss of refrigerant dosage) or it was manufactured with a major defect.[13] Although the exact cause of failure had not been determined at the time of report preparation, loss of lubrication was suspect. The premature failure of compressor #3 prevented a 'cold start' test from being conducted (see Figure 5.1, Cold Start Test on Compressor #2).




KEG WASTEWATER HEAT RECOVERY SYSTEM

HEAT PUMP PERFORMANCE TEST (COLD START)

(NOVEMBER 7, 1984)

FIGURE 5.1

| MONITORED PERIOD | WHRS MODE OF OPERATION | $Q_{HRS}$ ($MJ/10^4L$) | $E_{TOT}$ ($MJ/10^4L$) | $COP_{SYS}$ $\left\langle \frac{Q_{HRS}}{E_{TOT}} \right\rangle$ | $\Delta Q_{DHW}$ ($MJ/10^4L$) | $COP_{EFF}$ $\left\langle \frac{\Delta Q_{DHW}}{E_{TOT}} \right\rangle$ | PROJECTED ANNUAL ENERGY COST SAVING |
|---|---|---|---|---|---|---|---|
| Feb 15 - Mar 5 | 'Heat Exchanger Only' | 495 | 53 | 9.3(1) | 522 | 9.8(1) | $480 |
| Mar 15 - Mar 25 | 'Heat Pump Only' | 1 015 | 289 | 3.5 | 1 604 | 5.6 | $1 180 |
| Mar 26 - Mar 29 | 'Heat Exchanger and Heat Pump'(3) | 1 065 | 312(2) | 3.4 | 1 790 | 5.7 | $1 340 |

(1) This is not conventional terminology for expressing the performance of a passive heat exchanger.

(2) Increased electrical consumption foreshadows failure of compressor No. 3.

(3) Not considered a representative test for the combined mode of operation.

TABLE 5.1

## CONCLUSIONS & RECOMMENDATIONS

It was demonstrated that energy recovery from restaurant wastewater is technically feasible. However, the commercialization of such technology is going to require more than one site-specific demonstration project as a catalyst. Research and development must be continued, preferably with the direct involvement of a private sector company, if commercialization of the technology is to be a reality in Canada.

The measured heat recovery capacity of the heat pump centered wastewater heat recovery system fell short of the performance predicted by the design study, although the coefficient of performance (COP) measured for both the heat pump and the system met expectations. An energy cost saving of $1,300 per year was demonstrated, compared to the saving of $3,000 per year that had been anticipated by the designer. The associated economic constraint imposed on developing the technology is obvious. The cost of an installed heat recovery package should fall within $4 000 to $6 000 for the payback on the conservation investment to be acceptable to a restaurant owner.

As the project demonstrated, the cost benefit of heat pump systems will be very dependent on reliability. There is also a major difference between the application of heat pump technology for restaurant wastewater heat recovery and, for example, a large industrial process. While restaurant systems will range from 10 to 30 kW, an installation designed to recover heat from a pulp and paper process would be orders of magnitude larger. The large industrial installations will always be 'engineered' systems, able to absorb the design fee as a component of the system cost. This is just not economic for small capacity heat pump systems. For restaurant wastewater heat pump heat recovery to be effective, the system must become an off-the-shelf package. Maintenance should be handled through a service contract, as the operation of heat recovery equipment may not be well understood by restaurant staff.

Obviously there are major research and development steps that must be taken before heat pump centered, wastewater heat recovery technology will penetrate the Canadian restaurant and foodservice industry. The emphasis of future work should be directed towards the long term reliability and maintenance of the heat recovery system. The life expectancy of the heat pump compressor and fouling of heat exchange surfaces are a particular concern.

## CONCLUSIONS & RECOMMENDATIONS

It is recommended that the management of the Garry Street Keg continue to operate their wastewater heat recovery system. Although the economic picture does not suggest that similar systems should be installed in Keg restaurants across the country, there is a positive cost benefit to maintaining the existing system. The system should be modified to allow dishwater to pass through the passive heat exchanger before mixing with the cooler water in the holding tank. If evaporator fouling proves to be a maintenance burden, the heat pump should be used to only recover heat from the condenser cooling water discharged from the restaurant refrigeration equipment.

In conclusion, the Keg restaurants have demonstrated another Canadian first -- and there is no foreseen reason why it should be the last.

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11. 'Food Service and Hospitality' Magazine, September 1981, pp. 64.

12. Anon., 'Project on Restaurant Energy Performance - Interim Report', (National Restaurant Association, One Park Place/Suite 2000, Chicago, Illinois 60601) 1984.

13. Childs, W.S., 'Reliability Considerations for Heat Pump and Air-Conditioning Compressors'. ASHRAE Transactions 1983, V. 89, Pt. 1.

## APPENDIX 1

Photographs

Photo A1 - Keg Restaurant
Gary Street, Winnipeg

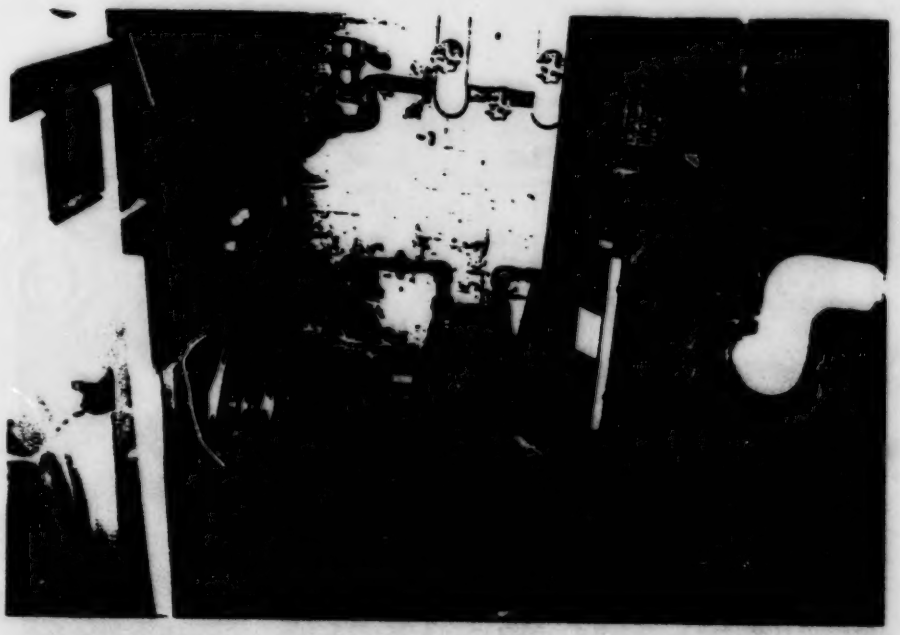


Photo A2 - Water and Gas Meter Installed on Inlet Lines to DHW Tank and Boiler

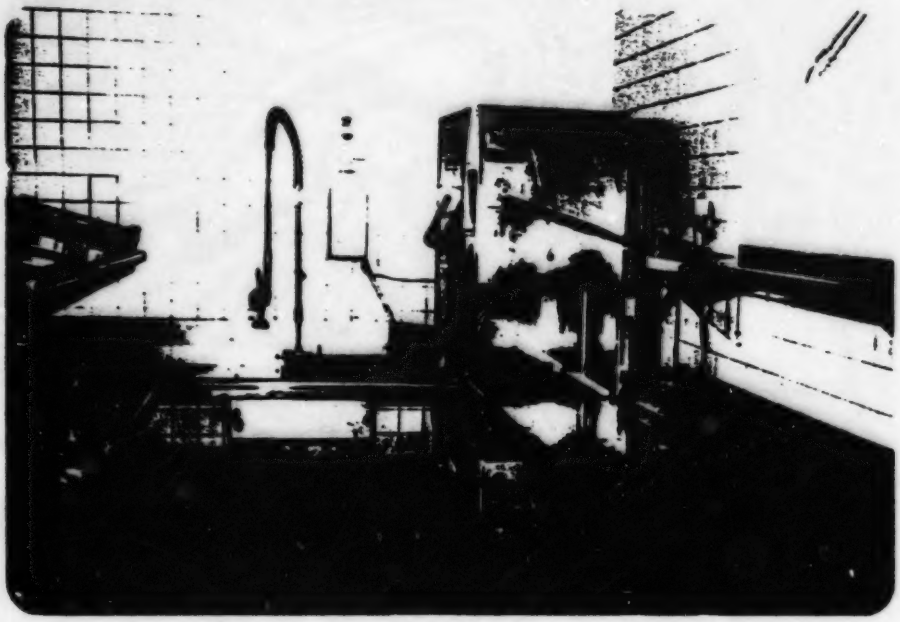


Photo A3 - Keg Dishwasher

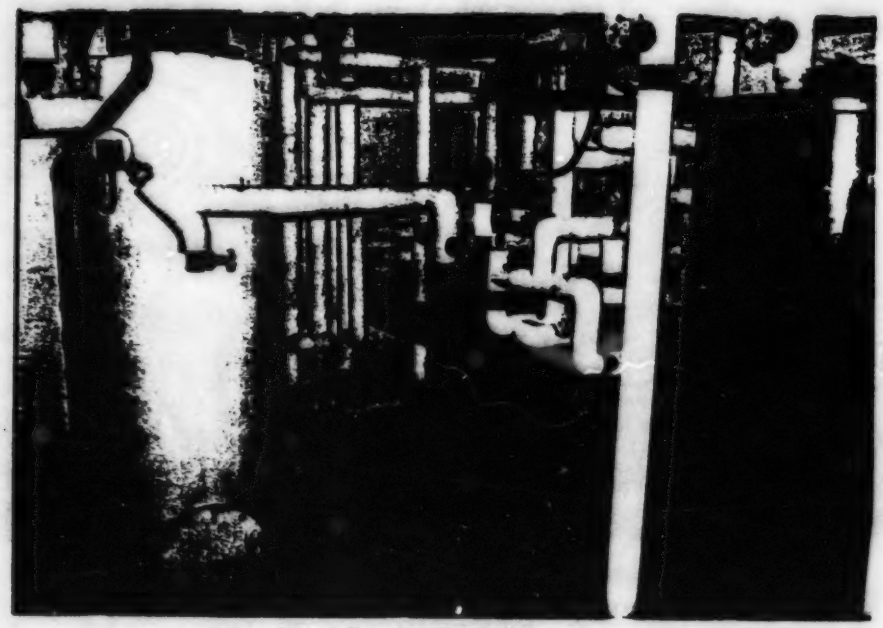


Photo A4 - WHRS

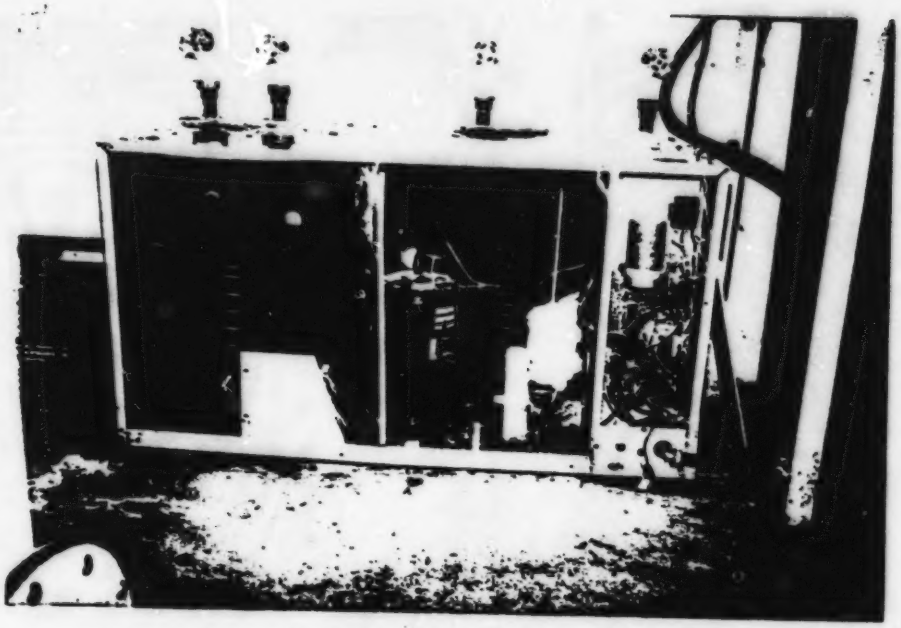


Photo A5 - Rotary Compressor Heat Pump

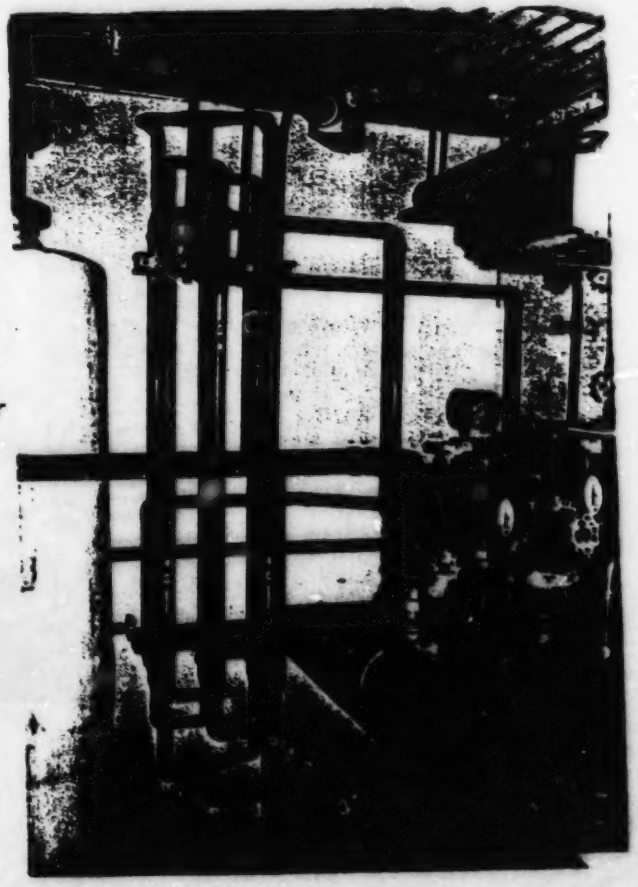


Photo A6
Double Wall Water-to-Water heat exchanger (before insulating)

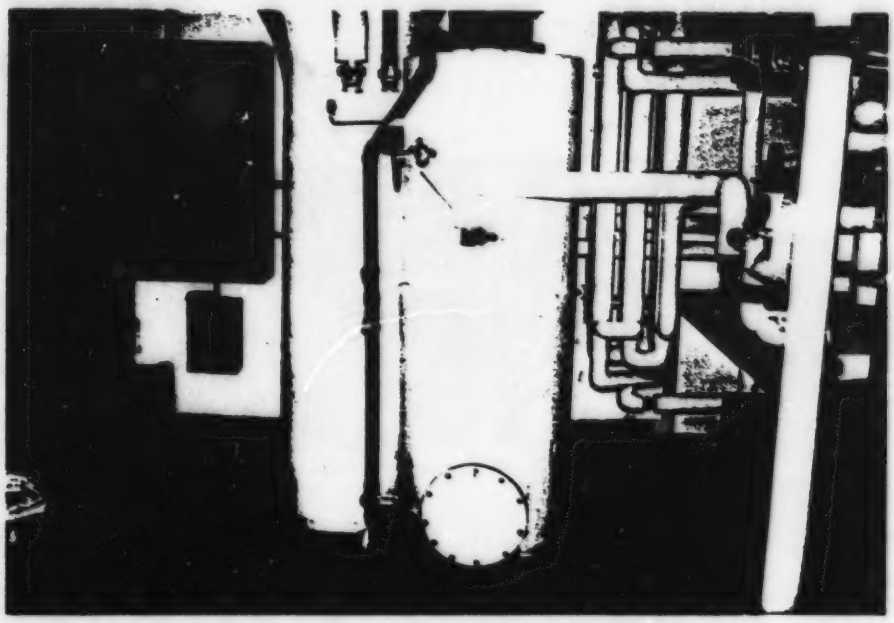


Photo A7 - Wastewater receiving tank (660 L), insulated heat exchanger, control cabinet and energy meter.

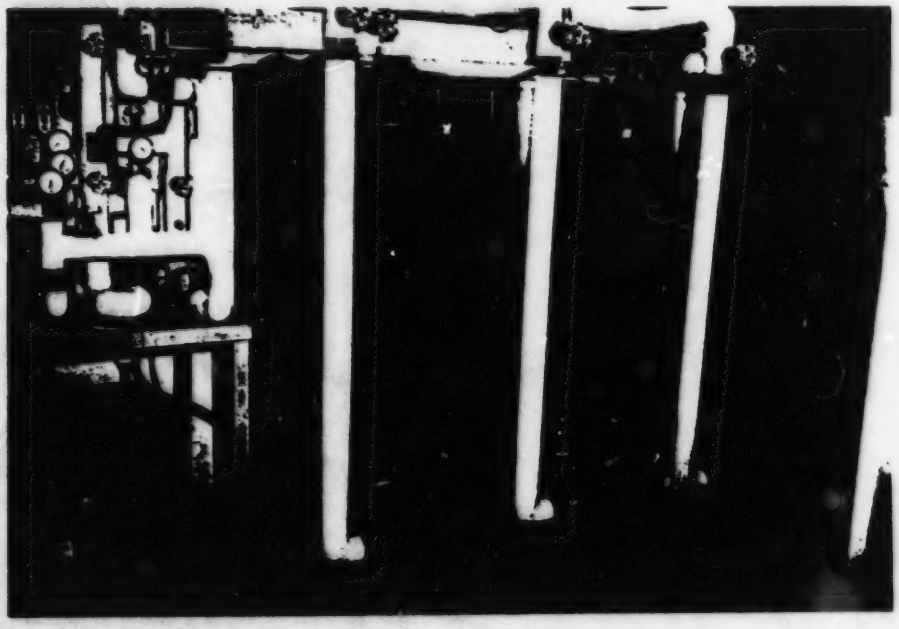


Photo A8 - Domestic Water Preheat Tanks (1.350 L)

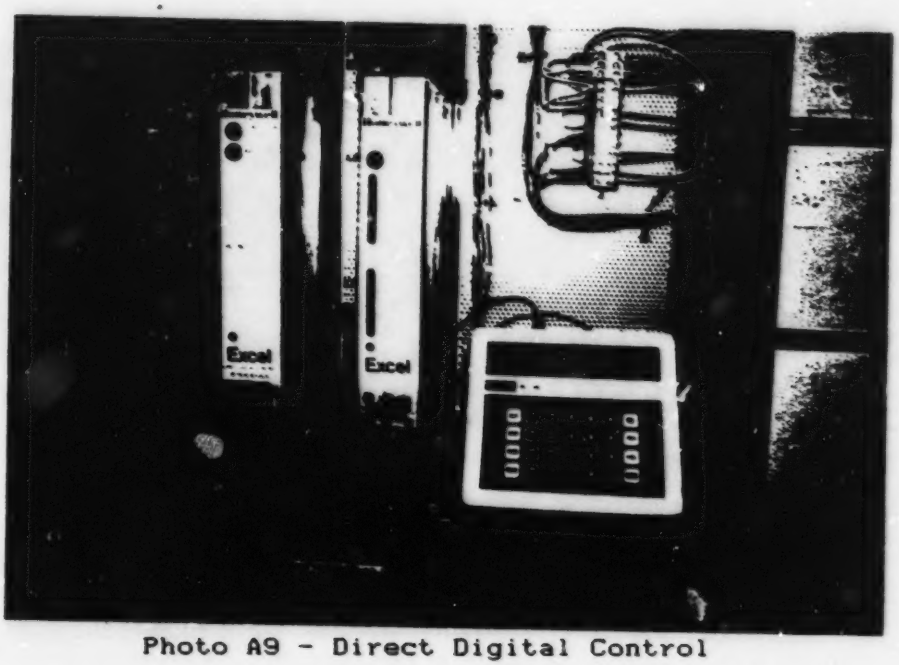



Photo A9 - Direct Digital Control

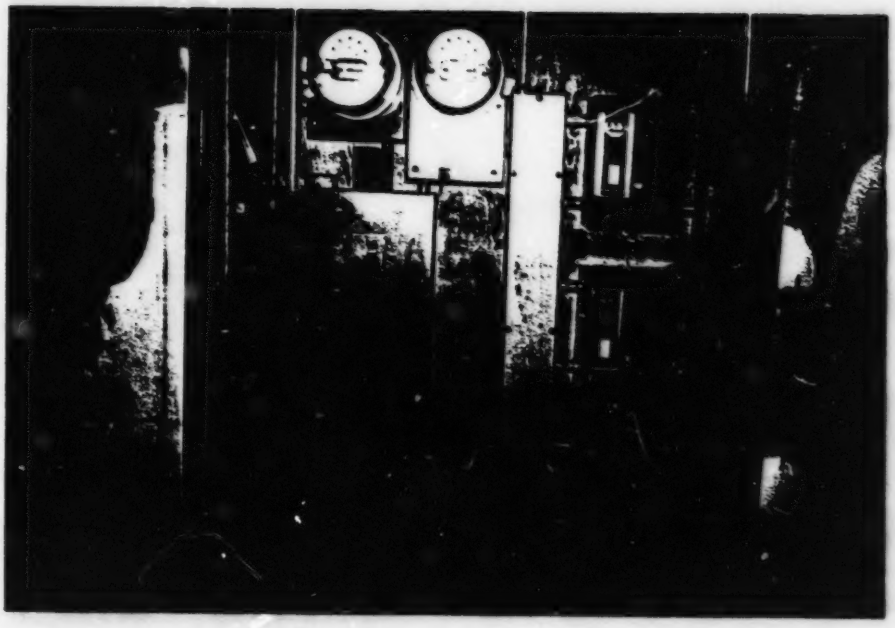


Photo A10 - Electrical Service for WHRS and kWh meters for heat pump and water pumps

Photo A11 - Plugged Impeller on Wastewater Pump

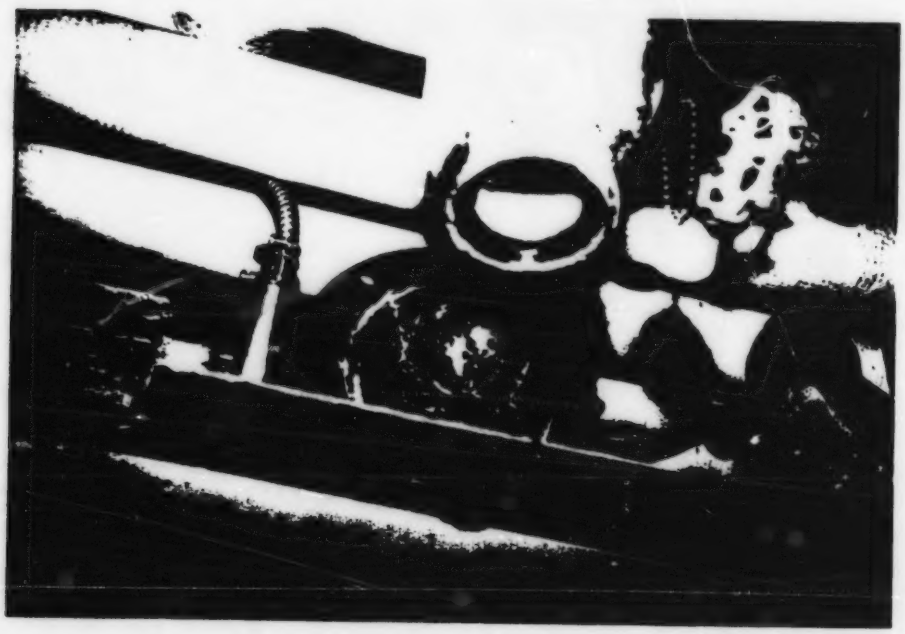


Photo A12 - Sludge Build-up in Wastewater Circuit

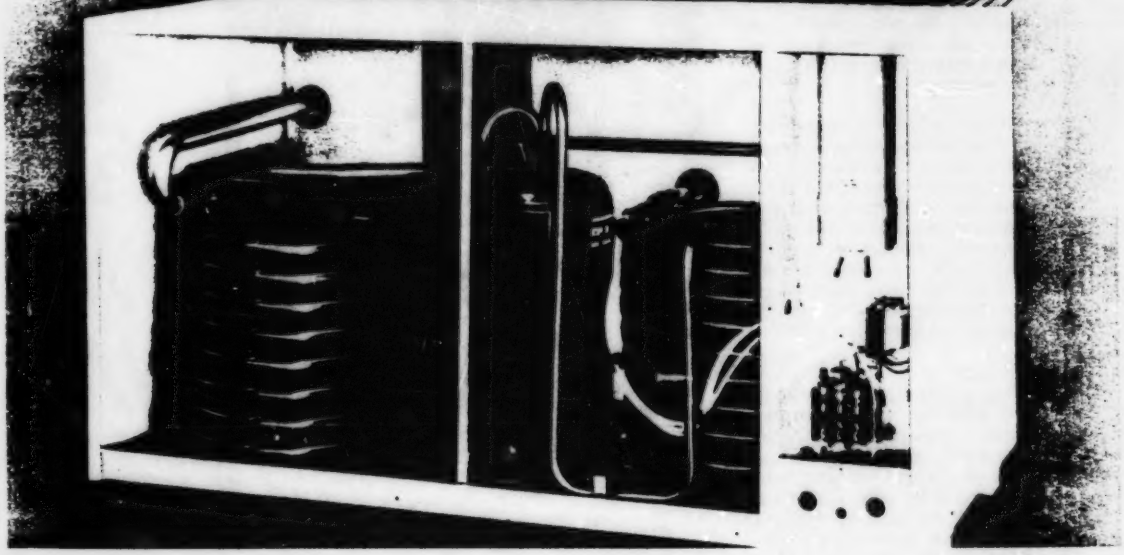
## APPENDIX 2

Heat Pump Specifications and

Performance Data

# The FEDDERS Solar/Co...

Sunrise for the Solar Age



Here is the Fedders Carnot II Solar/Compression Furnace. It repositions solar radiation as the most cost-effective energy resource for many space and water heating systems...and offers a broad range of heat recovery applications.

Up to now, the wide variations in radiant insolation over the course of a year...and its unpredictable input over the course of a day imprisoned solar heating applications between an unrealistically high initial cost...and a disappointingly low solar fraction. The Solar/Compression Furnace changes all that.

Connected in series to a solar collector-and-storage subsystem, Carnot II transforms solar heating into a technically practical, economically advantageous system for the first time. It raises solar storage temperatures between 60°F and 110°F to usable levels by a high-efficiency vapor compression system, rather than by supplementary resistance heat or fossil fuel as direct solar systems do.

Solar heated source water pumped through a coaxial counter-flow evaporator discharges heat to the refrigerant which is then raised to a higher temperature in the compressor. From there, it passes into a coaxial condenser from which it discharges high temperature water into a closed loop serving a terminal heating unit or other load. Similar equipment has been described as a "non-reversible heat pump". It should be noted, however, that in addition to lacking a reversing valve, there is no defrost requirement and no defrost controls; that the unit operates indoors and is largely independent of outdoor conditions; that no outdoor thermostatic control of supplementary heat is required.

## A WIDE RANGE OF OPERATING CONDITIONS

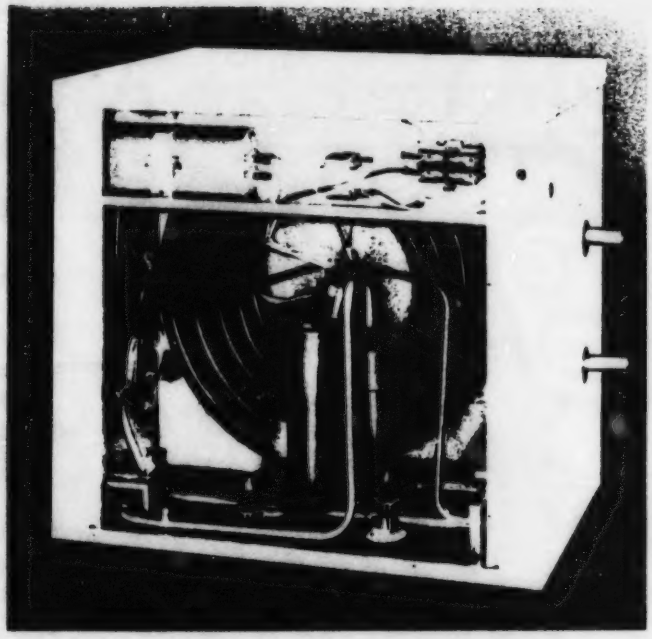
At high solar source temperatures when the Solar Compression Furnace operates at elevated evaporator temperatures, its efficiency increases. In contrast to the operation of conventional compression cycles, its watthour input actually declines. This reflects unique performance characteristics of the Carnot II's rotary compressor.

At low source temperatures, when stored energy has not been replenished due to extended overcast conditions, the Solar Compression Furnace resorts to groundwater as source energy. At an entering temperature of 55°F, groundwater from wells, ground grids or city mains permits the Solar/Compression Furnace to generate hot water with a COP of 4.2.

This defines the minimum operating efficiency of the unit and permits it to dispense completely with such back-up energy sources as resistance heat and fossil fuels which rob direct solar systems of efficiency. In systems properly sized to local levels of insolation, use of groundwater should be of brief duration and impose no significant cost penalty at this unusually high minimum efficiency. The absence of a regular energy source as a back-up will make the Solar Compression Furnace compatible to utility companies which display concern about the extra load that direct solar installations may impose during their peak operating conditions.

## LOW WORKING TEMPERATURES

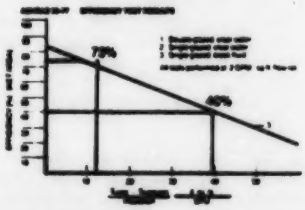
Carnot II's ability to use solar storage temperatures between 60°F and 110°F as a source without the aid of purchased thermal energy confers other important benefits on contractors and owners. Operation of solar subsystems at these low temperatures assures maximum absorption of radiant energy.



| | | | | | | |
|---|---|---|---|---|---|---|
| | | | | | | |
| | 28.8 | 16.2 | 10.8 | 4.1 | 0.2 | 0.0 |
| | | 39.4 | 28.1 | 17.5 | 9.3 | 3.1 |

This is apparent in standard certified ratings for solar collectors which list daily BTU output at six entering temperatures of heat transfer material as it is pumped into the collector from storage. At inlet temperatures of 75°F and 110°F, which approximate the Solar Compression Furnace's operating range, collector output is from 100% to 300% greater than at the higher collector temperatures required by "direct" solar systems.

This advantage can be seen also in collector efficiency curves which plot a ratio of the fluid-ambient T/insolation against collector efficiency on the vertical axis. The closer incoming fluid is to outdoor winter temperatures, the more solar energy is absorbed. The difference in efficiency is inversely reflected in square feet of collector surface required: the lower the efficiency, the greater the collector area. The Solar Compression Furance can therefore reduce solar collector costs to a significant degree.




## HEAT RECOVERY CAPABILITY

The ability of the Solar Compression Furnace to operate efficiently at high evaporating temperatures equips it for a wide variety of heat recovery applications. In commercial and industrial systems where waste process heat can supply a no-cost continuous supply of source energy, the

Solar Compression Furnace can make an important bottom-line contribution in business as varied as fast food restaurants and casting foundries.

If heating or cooling of process liquids is required, such liquids can be pumped directly through the evaporator or condenser which are both equipped with cupro nickel inner tubes in all models of the compression furnace. Brines, degreasing solutions and a variety of mildly corrosive process liquids can be handled directly without the losses associated with intermediate heat exchangers.

Heat recovery with the Solar Compression Furnace can be far less costly and far more flexible in many installations where air-to-air systems are the only alternative. Massive ducts, high power blowers, heat wheels and other exchangers required in air-to-air exchange can be avoided by a ¾-inch type "L" copper pipe line and a compact Solar Compression Furnace which can be installed remotely wherever needed.

## THE ROTARY COMPRESSOR

The high COP's of the Solar Compression Furnace reflects performance of its rolling piston rotary compressor. This has been confirmed by independent researchers in a Federally financed DOE contract.*

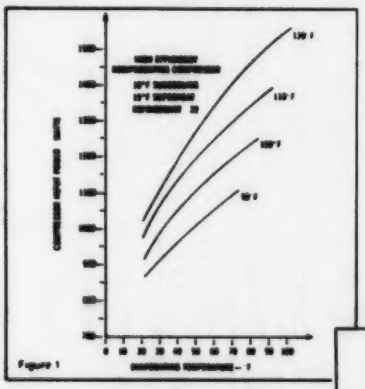
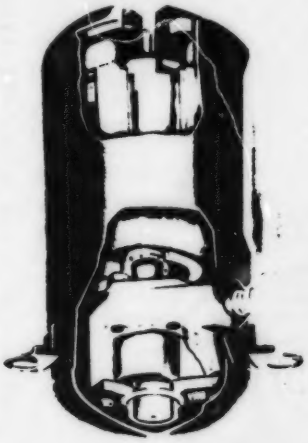


Figure 1



In the graph above, typical reciprocating compressors display a sharp increase in power input for different condensing temperatures as evaporating temperatures increase. Under identical conditions, power input for the Solar Compression Furnace's rotary compressor in the lower graph declines as evaporating temperatures increase. This difference in energy consumption which can amount to as much as 40% higher for reciprocating compressors, accounts for Carnot II's remarkable operating efficiency.

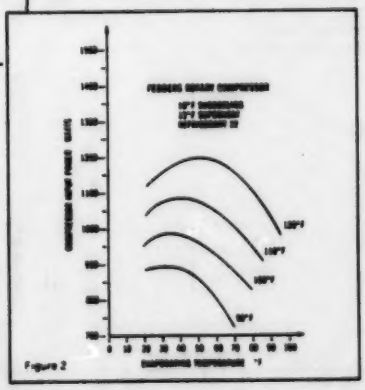


Figure 2

*Write for copy of technical paper.

## EQUIPMENT

The Solar Compression Carnot II Furnace is available in four capacities which are denominated in heating BTU's. They are supplied complete, ready to operate when water and power lines are connected.

They are all equipped with rotary compressors with internal overloads and counter-flow coaxial condensers and evaporators. To meet a range of operating conditions, the double wall inner tube is fabricated of corrosion resistant cupro-nickel. Automatic thermal expansion valves are factory installed. Contactors, relays and other components are factory-wired into the control compartment. The entire assembly rests on a heavy gauge steel base with structural support members and is housed in a baked enamel steel cabinet.

## UNIT RATINGS

The ratings in the following tables refer only to the unit output of hot water at the indicated entering evaporator water. They do not include the significant solar fraction when collector temperatures exceed the required condensing temperature and solar radiated water moves directly to the condenser bypassing the vapor compression cycle.

**% GAIN IN HEATING OUTPUT WHEN EVAPORATOR FLOW RATE OF SOCF040-700 IS INCREASED***

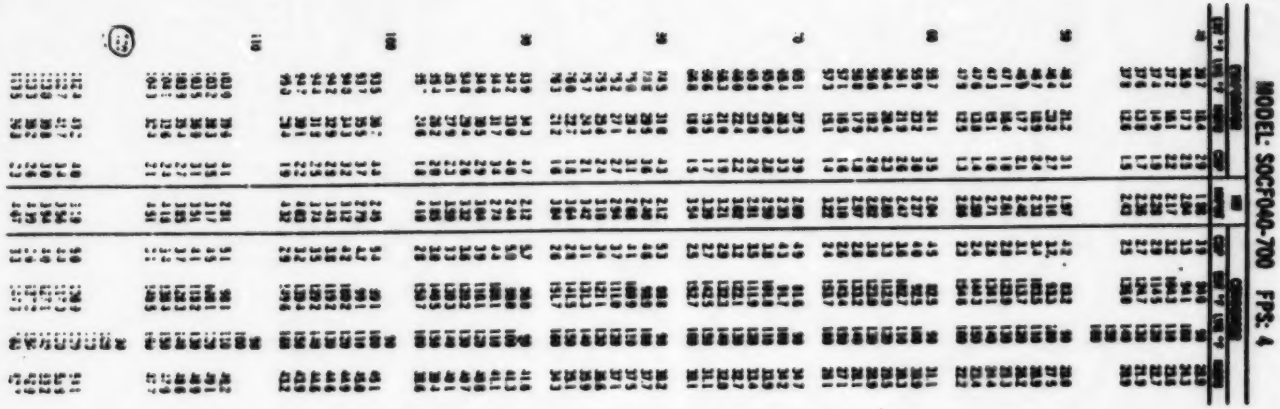
| Entering Evap. Water Temp. °F | From 2 FPS (5.7 gpm) to 6 FPS (17.1 gpm) | From 2 FPS (5.7 gpm) to 4 FPS (11.4 gpm) | From 4 FPS (11.4 gpm) to 6 FPS (17.1 gpm) |
|---|---|---|---|
| 40° | 11.2 | 6.5 | 4.4 |
| 50° | 12.3 | 6.8 | 5.0 |
| 60° | 13.5 | 7.7 | 5.4 |
| 70° | 14.9 | 8.3 | 5.9 |
| 80° | 16.5 | 9.7 | 6.2 |
| 90° | 18.2 | 10.8 | 6.6 |
| 100° | 18.7 | 10.3 | 7.7 |
| 110° | 18.3 | 9.8 | 7.7 |
| 120° | 19.0 | 10.0 | 8.1 |

* Average for all leaving condenser water temperatures. 90°F to 160°F Condenser flow rate is not changed.

**MODEL: SOCF040-700 FPS: 2**

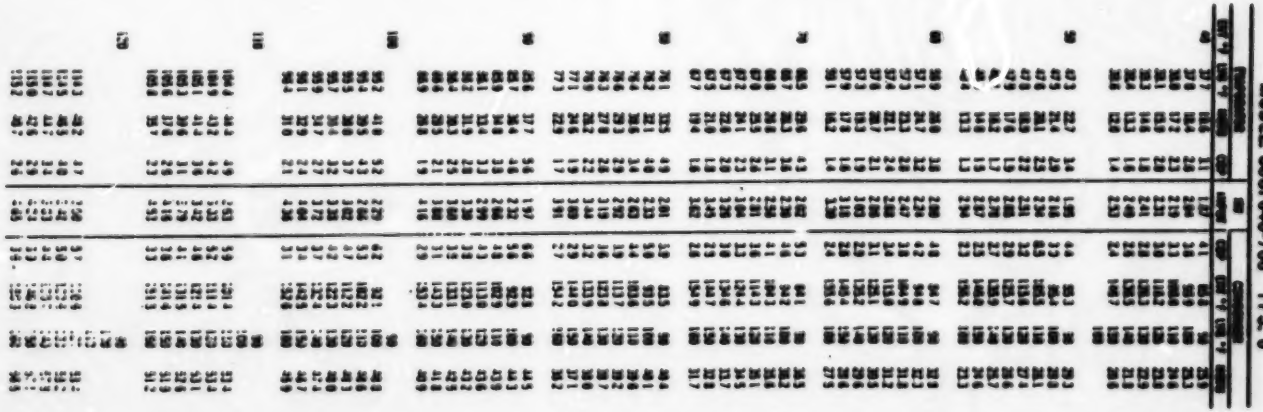
| EVAPORATOR | | | | KW | CONDENSER | | | |
|---|---|---|---|---|---|---|---|---|
| EWT °F | LWT °F | MBHR | GPM | INPUT | GPM | EWT °F | LWT °F | MBHR |
| 40 | 348 | 171 | 27 | 185 | 37 | 787 | 90 | 235 |
| | 343 | 162 | 24 | 200 | 34 | 869 | 100 | 250 |
| | 347 | 151 | 21 | 212 | 31 | 992 | 110 | 323 |
| | 351 | 140 | 18 | 221 | 28 | 1096 | 120 | 316 |
| | 355 | 129 | 16 | 229 | 26 | 1200 | 130 | 307 |
| | 359 | 118 | 15 | 234 | 25 | 1304 | 140 | 198 |
| | | | | | | | 150 | |
| | | | | | | | 160 | |
| 50 | 431 | 197 | 30 | 195 | 40 | 773 | 90 | 264 |
| | 434 | 187 | 26 | 212 | 36 | 875 | 100 | 259 |
| | 438 | 175 | 22 | 223 | 32 | 978 | 110 | 253 |
| | 442 | 164 | 20 | 241 | 30 | 1081 | 120 | 246 |
| | 446 | 153 | 18 | 253 | 28 | 1184 | 130 | 239 |
| | 450 | 142 | 16 | 262 | 26 | 1288 | 140 | 231 |
| | 454 | 129 | 14 | 270 | 24 | 1393 | 150 | 221 |
| | 459 | 115 | 12 | 273 | 22 | 1491 | 160 | 209 |
| 60 | 521 | 223 | 32 | 204 | 42 | 759 | 90 | 282 |
| | 525 | 212 | 28 | 224 | 38 | 861 | 100 | 289 |
| | 529 | 200 | 24 | 242 | 34 | 963 | 110 | 283 |
| | 533 | 188 | 21 | 259 | 31 | 1066 | 120 | 278 |
| | 537 | 178 | 19 | 274 | 29 | 1169 | 130 | 272 |
| | 541 | 166 | 17 | 287 | 27 | 1272 | 140 | 264 |
| | 546 | 154 | 15 | 298 | 25 | 1377 | 150 | 255 |
| | 551 | 138 | 13 | 306 | 23 | 1482 | 160 | 243 |
| 70 | 612 | 249 | 35 | 211 | 45 | 745 | 90 | 321 |
| | 616 | 238 | 30 | 233 | 40 | 847 | 100 | 314 |
| | 620 | 227 | 26 | 254 | 36 | 948 | 110 | 314 |
| | 624 | 216 | 23 | 274 | 33 | 1050 | 120 | 310 |
| | 628 | 204 | 20 | 293 | 30 | 1153 | 130 | 304 |
| | 632 | 192 | 18 | 310 | 28 | 1256 | 140 | 297 |
| | 637 | 179 | 16 | 325 | 26 | 1360 | 150 | 290 |
| | 641 | 167 | 14 | 340 | 24 | 1464 | 160 | 282 |
| 80 | 702 | 277 | 38 | 216 | 48 | 731 | 90 | 351 |
| | 706 | 266 | 32 | 241 | 42 | 832 | 100 | 344 |
| | 710 | 255 | 29 | 265 | 38 | 933 | 110 | 345 |
| | 714 | 243 | 25 | 288 | 35 | 1035 | 120 | 342 |
| | 718 | 231 | 22 | 310 | 32 | 1137 | 130 | 337 |
| | 723 | 219 | 19 | 330 | 29 | 1240 | 140 | 332 |
| | 727 | 206 | 17 | 350 | 27 | 1343 | 150 | 325 |
| | 732 | 192 | 15 | 368 | 25 | 1447 | 160 | 315 |
| 90 | 791 | 309 | 41 | 228 | 51 | 715 | 90 | 383 |
| | 795 | 297 | 35 | 247 | 45 | 816 | 100 | 381 |
| | 799 | 285 | 31 | 274 | 41 | 917 | 110 | 379 |
| | 803 | 274 | 27 | 300 | 37 | 1018 | 120 | 374 |
| | 808 | 261 | 24 | 325 | 34 | 1120 | 130 | 372 |
| | 812 | 249 | 21 | 349 | 31 | 1222 | 140 | 368 |
| | 817 | 234 | 18 | 372 | 29 | 1325 | 150 | 362 |
| | 822 | 220 | 16 | 393 | 25 | 1429 | 160 | 355 |
| 100 | 884 | 331 | 39 | 250 | 49 | 739 | 90 | 416 |
| | 888 | 315 | 33 | 270 | 43 | 840 | 100 | 414 |
| | 892 | 307 | 29 | 300 | 39 | 1001 | 120 | 412 |
| | 896 | 294 | 26 | 327 | 36 | 1102 | 130 | 409 |
| | 901 | 281 | 23 | 354 | 33 | 1204 | 140 | 405 |
| | 906 | 266 | 20 | 391 | 30 | 1307 | 150 | 400 |
| | 911 | 253 | 18 | 417 | 29 | 1409 | 160 | 395 |
| 110 | | | | | | | 90 | |
| | | | | | | | 100 | |
| | 975 | 365 | 37 | 284 | 47 | 881 | 110 | 452 |
| | 979 | 343 | 32 | 316 | 42 | 982 | 120 | 451 |
| | 983 | 330 | 29 | 347 | 38 | 1083 | 130 | 449 |
| | 989 | 316 | 25 | 379 | 35 | 1185 | 140 | 445 |
| | 994 | 302 | 22 | 407 | 32 | 1287 | 150 | 441 |
| | 999 | 288 | 19 | 437 | 29 | 1389 | 160 | 435 |
| 120 | | | | | | | 90 | |
| | | | | | | | 100 | |
| | | | | | | | 110 | |
| | 1067 | 376 | 34 | 355 | 44 | 945 | 120 | 484 |
| | 1070 | 365 | 31 | 355 | 41 | 1045 | 130 | 486 |
| | 1075 | 354 | 27 | 398 | 37 | 1145 | 140 | 486 |
| | 1082 | 335 | 23 | 421 | 33 | 1245 | 150 | 478 |
| | 1085 | 326 | 21 | 454 | 31 | 1346 | 160 | 471 |

2 FEET/SEC. EVAP. GPM= 5.7 ΔPSI = 1.2; FT.HEAD = 2.8; COND. GPM = 4.2; ΔPSI = 1.3; FT. HEAD = 2.9



MODEL: SOCF040-700 FPS: 4

4 FEET/SEC. EVAP: GPM= 11.4 ΔPSI = 4.0; FT.HEAD = 9.2; COND: GPM = 8.3; ΔPSI = 4.2; FT. HEAD = 9.7



MODEL: SOCF040-700 FPS: 6

6 FEET/SEC. EVAP: GPM= 17.1 ΔPSI = 8.2; FT.HEAD = 18.9; COND: GPM = 12.5; ΔPSI = 8.5; FT. HEAD = 19.6

## APPENDIX 3

### Heat Exchanger Specifications

HERE'S ANOTHER BRIGHT IDEA FOR THE SOLAR INDUSTRY FROM noranda

"d-a"
heat exchanger
AVAILABLE WITH LEAK DETECTION

UL LISTED

- GREATER EXPOSED SURFACE AREA
  (Photo A) ... higher heat transfer from same size unit or receive same thermal recovery from a smaller size.
- LEAK DETECTION FEATURE
  ... because the leak passages terminate outside the heat exchanger for prompt detection.
- 2 WEEK DELIVERY
  ... standard heat exchangers.
  Other sizes made to order.

STANDARD SIZES

noranda

NORANDA METAL INDUSTRIES, INC.
Forge-Fin Division
Prospect Drive
Newtown, CT 06470
(203) 426-4431

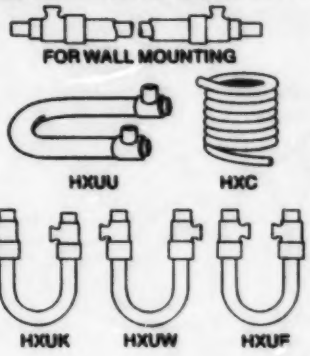
## Applications:

Noranda "d-a" Heat Exchangers provide superior performance in most low energy heat recovery/transfer situations. Standard and custom designed heat exchangers are made with and without double-augmentation and leak-detection. Independent studies have shown double-augmentation to increase heat transfer up to 150%. It should also be noted that where leak-detection is not required, heat exchangers perform with increased efficiency.

Media for which Noranda Heat Exchangers are used include refrigerants, oils, gray water, air, and all kinds of solar fluids. Performance curves for other than the glycol/water and water provided, are available upon request.

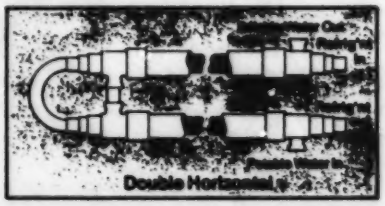
## Configuration Possibilities:

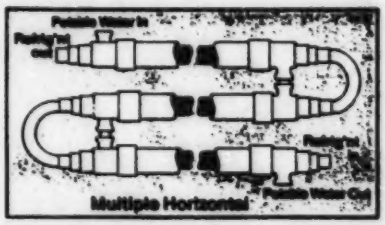
Noranda Heat Exchangers can be altered in shape to fit special applications and to increase efficiency. Heat exchangers can be bent or coiled, and made for wall-mounting. Below are our most common adaptations.

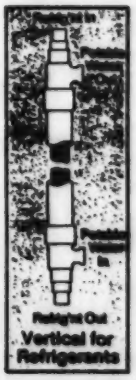


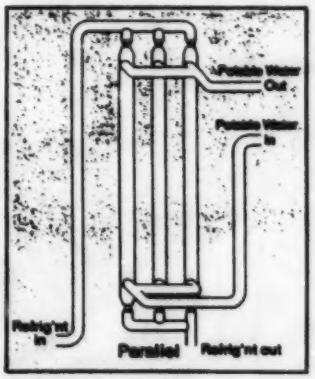

## Hook-up & Variations:

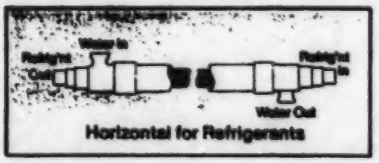
Utilizing the convenient "K" model (see inside), Noranda Heat Exchangers can be installed in-line in any number of manners:

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**NORANDA METAL INDUSTRIES, INC.**
Forge-Fin Division
Prospect Drive, Newtown, CT 06470
(203) 426-4431

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HERE'S ANOTHER BRIGHT IDEA IN HEAT RECOVERY FROM noranda

REPLY TO HEAT RECOVERY PRODUCT INQUIRY

E. J. PERACI & ASSOCIATES
JUN 18 1984
FILE No.
Date: 6/7/84
REMARKS

FOR: Attn: Don Fisher
E.J. Peraci & Assoc.
200 - 330 St Mary Ave
Winnipeg, Manitoba R3C 3Z5

EF
DF.

| | Hot Stream (tube) | Cold Stream (shell) |
|---|---|---|
| Type Fluid | water | water |
| Flow Rate, GPM | 8 | 8 |
| Temp In °F | 120 | 60 |
| Temp Out °F | 90 | 90 |
| ▲ P, PSI | 1.5 | 2.3 |

Heat Transfer BTU/HR: 120,000

Calculated LMTD °F: 30

SUGGESTED PRODUCT: (6) HX 148875 LDDA x 4 feet
- tubeside connected 3 pieces in series to form group with two groups in parallel
- shellside connected in series

Delivery from stock

* Caution: This data was not furnished. In order that sizing computations be made, this data was assumed by our Engineers. If our assumptions are not correct the sizing data given can be in error.

noranda

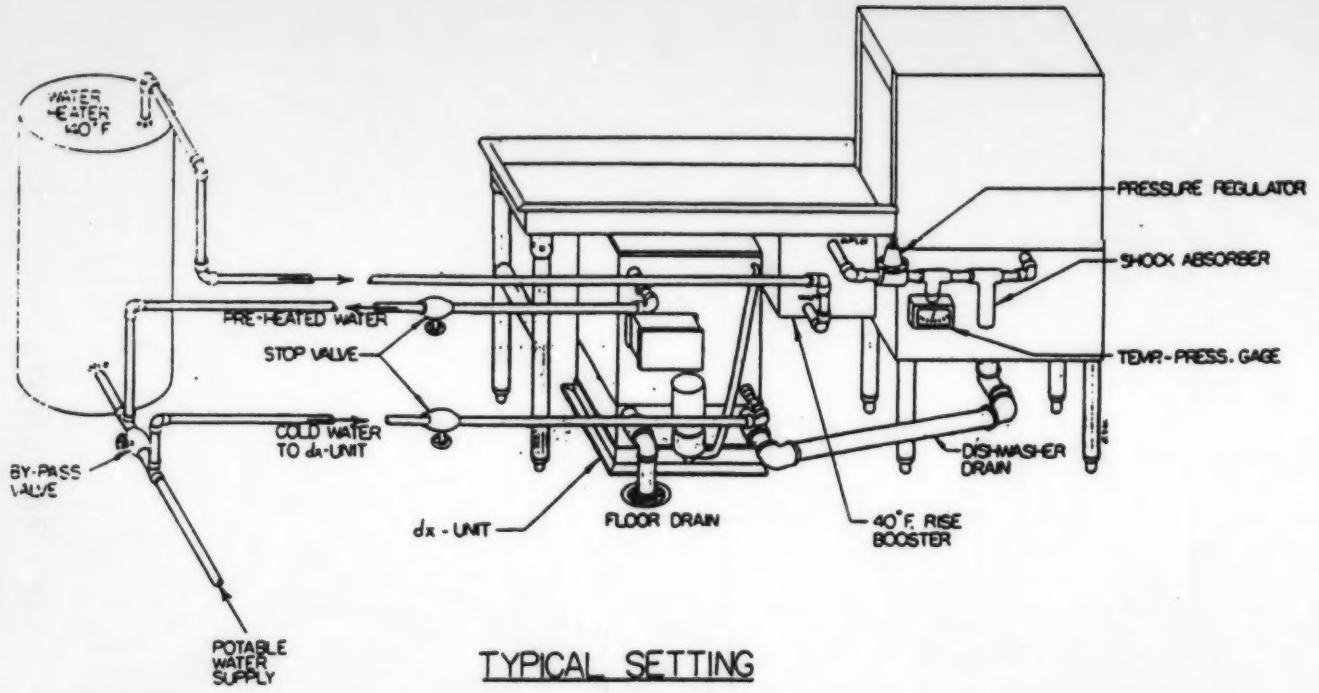
NORANDA METAL INDUSTRIES, Inc.

NORANDA METAL INDUSTRIES, INC.
27 Glen Road
Sandy Hook, CT 06482

## APPENDIX 4

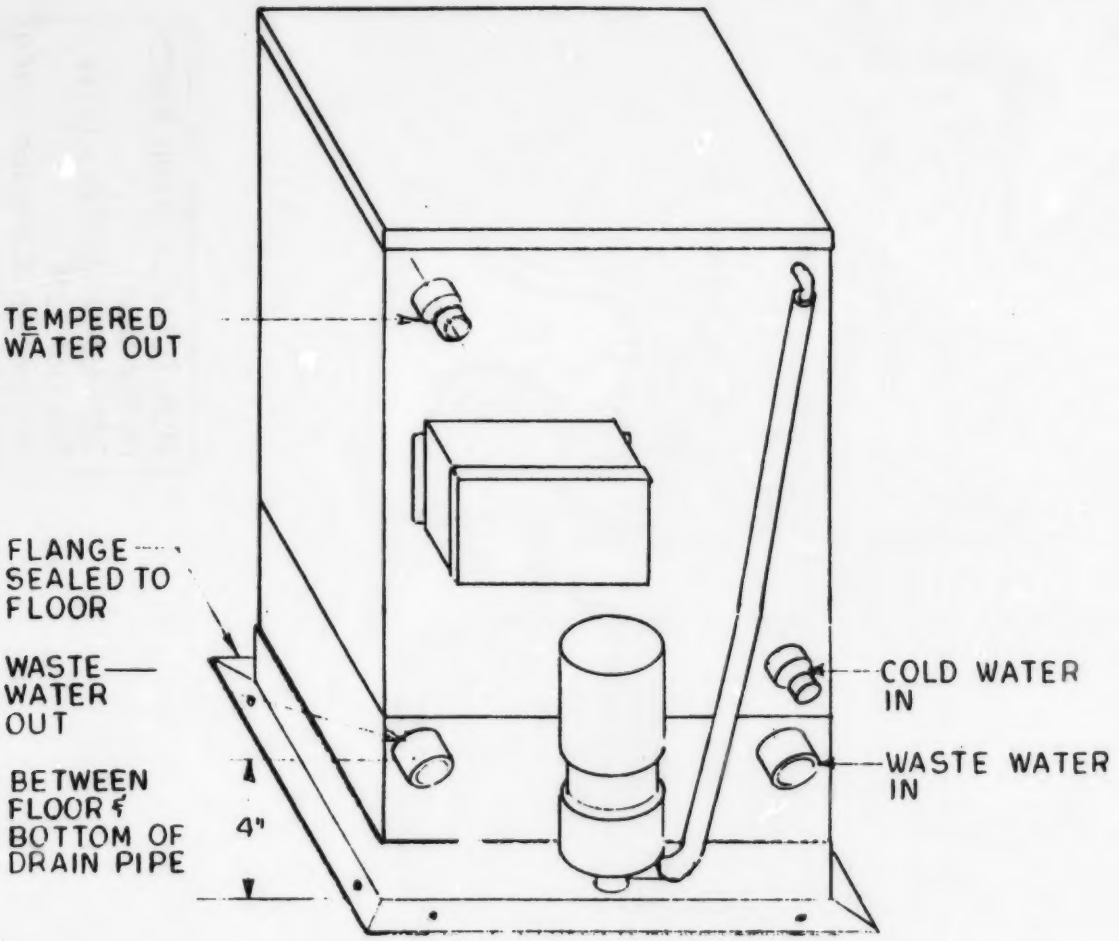
Add-On Heat Exchanger for

Commercial Dishwashers




TYPICAL SETTING

**MOLITOR INDUSTRIES INC.**
**dx Division**
**2829 South Santa Fe Drive**
**P.O. Box 1218**
**Englewood, Colorado 80150**
**303-789-2231**




DX - HEAT EXCHANGER
W/ S/S FLOOR FLANGE

12-30-83

# Molitor~dx

THE FOLLOWING FIGURES ARE AVERAGE DOLLAR SAVINGS FOR FOOD SERVICE OPERATIONS USING THE MOLI-DX HEAT EXCHANGER IN CONJUNCTION WITH THEIR DISHWASHER.

## - SINGLE RACK MACHINES -

| TYPE OF FUEL FOR WATER HEATER | ANNUAL SAVINGS WITH DX-2 HIGH TEMP MACHINE | LOW TEMP MACHINE |
|---|---|---|
| Electric ($.072/KW) | $ 1,714.00 | $ 1,419.00 |
| Oil ($1.28/Gal) | $ 1,020.00 | $ 752.00 |
| Propane ($.858/Gal) | $ 1,074.00 | $ 790.00 |
| Natural Gas ($.591/100 $Ft^3$) | $ 600.00 | $ 497.00 |

## - DOUBLE RACK MACHINES OR SMALL CONVEYORS -

| TYPE OF FUEL FOR WATER HEATER | ANNUAL SAVINGS WITH DX-4 HIGH TEMP MACHINE | LOW TEMP MACHINE |
|---|---|---|
| Electric ($.072/KW) | $ 4,690.00 | $ 3,689.00 |
| Oil ($1.28/Gal) | $ 2,654.00 | $ 1,956.00 |
| Propane ($.858/Gal) | $ 2,790.00 | $ 2,056.00 |
| Natural Gas ($.591/100 $Ft^3$) | $ 1,643.00 | $ 1,292.00 |

(Based on Average Colorado Fuel Costs).

12-30-83

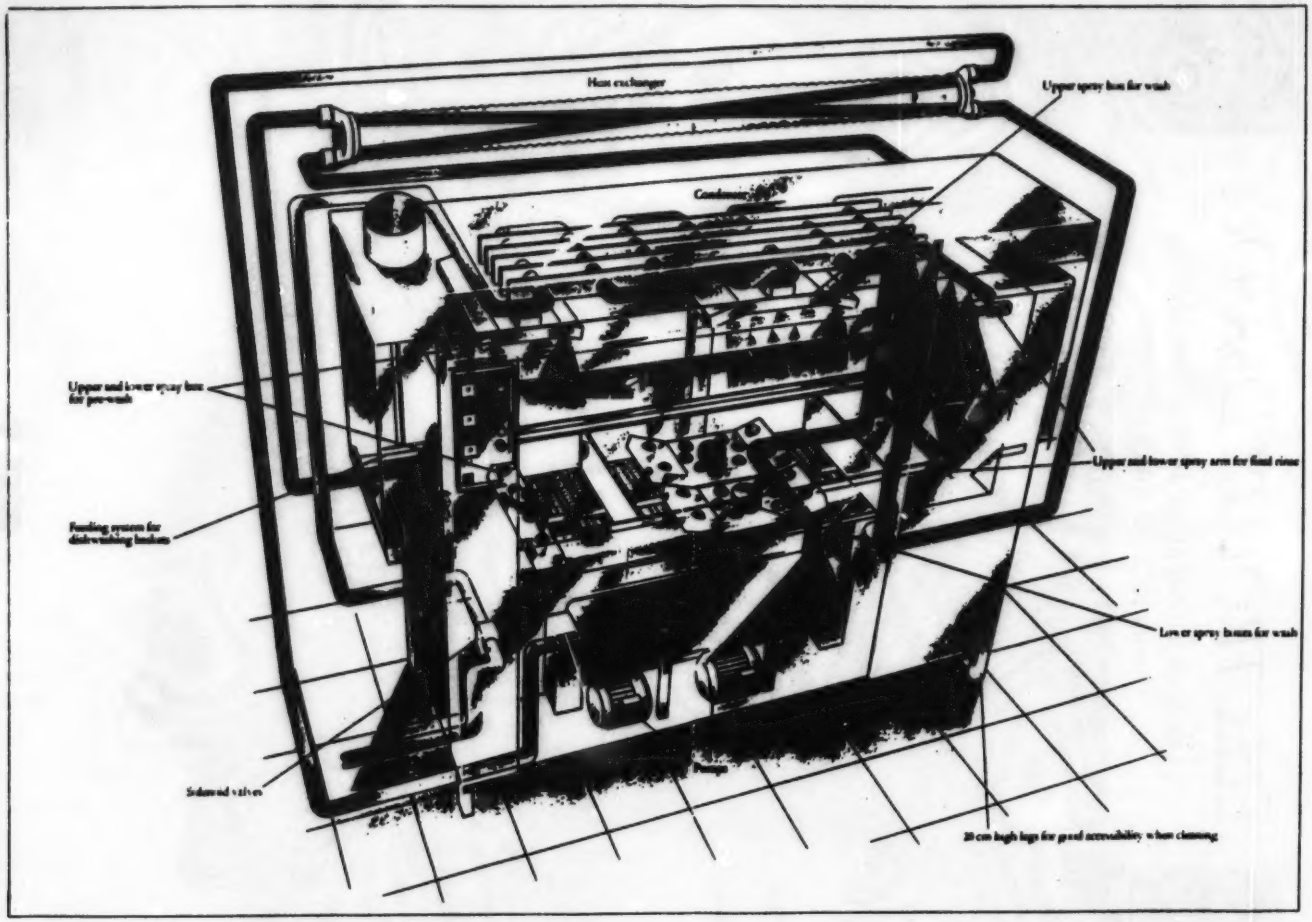
## APPENDIX 5

Swedish Dishwasher with Integral

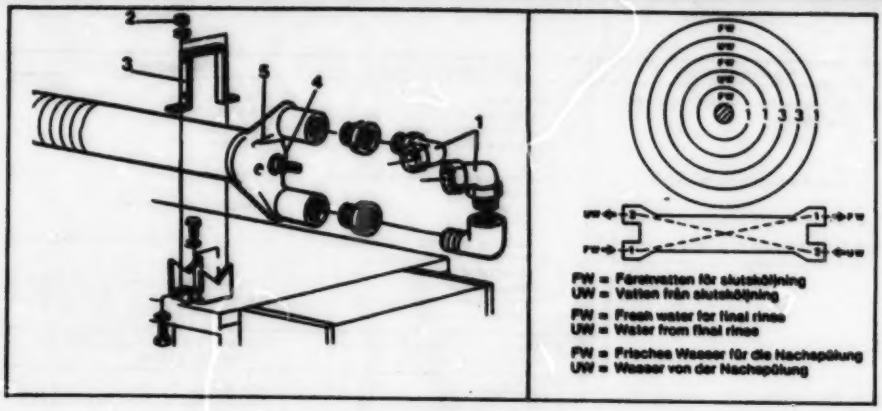
Wastewater Heat Exchanger

# ELECTROLUX D58/D68

*The professional dishwasher with a capacity of up to 3600 plates per hour.*


Heat exchanger
Upper spray bar for wash
Upper and lower spray bar for pre-wash
Feeding system for dishwashing products
Upper and lower spray arm for final rinse
Lower spray bars for wash
Solenoid valves
20 cm high legs for good accessibility when cleaning

| Fault symptom | Conceivable faults | Remedial action |
|---|---|---|
| 17 Temperature in chemical wash tank too high. | 1 Relay fault.<br>2 The thermostat is wrongly set.<br>3 The thermostat is faulty.<br>4 The overheating protector is faulty. | 1 Replace.<br>2 Adjust.<br>3 Replace.<br>4 Replace. |
| 18 Final rinse water not heated. | 1 The overheating protector has been triggered.<br>2 Relay fault. | 1 Reset.<br>2 Replace coil or relays. |
| 19 Temperature of final rinse too low. | 1 Booster not working, thermostat incorrectly set.<br>2 Thermostat is faulty.<br>3 Heating element is faulty.<br>4 Connection to element is faulty.<br>5 Water flow too high. | 1 Adjust. See instructions on page 3.4.<br>2 Replace.<br>3 Replace. See instructions on page 3.4.<br>4 Replace. See instructions on page 3.4.<br>5 Adjust the regulator valve. See instructions on page 3.8. |
| | 6 Temperature of water from heat exchanger too low. | 6 Clean heat exchanger, pump and connecting pipes. |
| 20 Temperature of final rinse too high. | 1 Relay fault.<br>2 The thermostat is wrongly set.<br>3 The thermostat is faulty.<br>4 The overheating protector is faulty. | 1 Replace.<br>2 Adjust. See instructions on page 3.3.<br>3 Replace.<br>4 Replace. |
| 21 Steam coming from openings in machine. | 1 Entrance/exit curtains incorrectly placed.<br>2 Condenser fan not working.<br>3 Air filter in front of fan blocked.<br>4 Timer on PC board incorrect.<br>5 PC board faulty.<br>6 Relay faulty.<br>7 Capacitor faulty. | 1 Correct. See instructions on page 3.18.<br>2 Replace fan, if it has burned out.<br>3 Replace filter.<br>4 Adjust. See instruction on page 2.3.<br>5 Replace PC board.<br>6 Replace relay R8 or R9. See Wiring Diagram.<br>7 Replace capacitor. |
| 23 Air from condenser unit is hot and humid. | 1 Cooling water temperature too high.<br>2 Cooling water control valve incorrectly adjusted.<br>3 Dirt in control valve.<br>4 Cooling water solenoid valve blocked.<br>5 The solenoid valve coil has burnt out.<br>6 Timer on PC board incorrect.<br>7 PC board faulty.<br>8 Relay faulty. | 1 Inform person responsible for water supply in building.<br>2 Adjust as shown in Description of Functions, page 3.10.<br>3 Clean valve.<br>4 Dismantle solenoid valve and clean. See instruction on page 3.5.<br>5 Replace the coil.<br>6 Adjust. See instruction on page 2.3.<br>7 Replace PC board.<br>8 Replace relay R8 or R9. See Wiring diagram. |




**Värmeväxlare**

| Se bild | Pos nr |
|---|---|
| 1 Lossa samtliga kopplingar i värmeväxlarnas båda ändar. Märk upp samtliga anslutningar och motsvarande rör. | 1 |
| 2 Skruva ur de två bakre skruvarna på klämbyglarna. | 2 |
| 3 Böj klämbyglarna framåt och lyft ur värmeväxlaren. | 3 |
| 4 Skruva bort centrummuttern i ena änden på värmeväxlaren. | 4 |
| 5 Knacka ur det gjutna ändstycket från rören med en gummiklubba. Märk upp ändstycke mot det yttre röret. | 5 |
| 6 Drag ur rören för rengöring. Märk upp rören i förhållande till varandra. | |
| 7 Rengör med borste och ett surt hålmedel. | |

Montera i omvänd ordning. Smörj in O-ringarna med vaselin. Om rören trots märkning blivit blandade, montera enligt siffrorna på skissen.

**Heat Exchanger**

| See fig. | Pos no |
|---|---|
| 1 Undo all connections at both ends of the heat exchanger. Mark all connections with their corresponding pipes. | 1 |
| 2 Unscrew the two rear screws on the binding clamps. | 2 |
| 3 Bend the binding clamps forwards and lift out the heat exchanger. | 3 |
| 4 Unscrew the centre nut at one end of the heat exchanger. | 4 |
| 5 Knock out the cast endpiece from the pipes with a rubber mallet. Mark the end-piece against the outer pipe. | 5 |
| 6 Remove the pipe for cleaning. Mark the pipes' position in relation to each other. | |
| 7 Clean using a brush and acid detergent. | |

Reassemble in reverse order to the above.
Grease the O-rings with Vaseline. Should the pipes despite marking become mixed up, reassemble according to the numbers on the diagram.

**Wärmeaustauscher**

| Siehe Abb. | Pos nr. |
|---|---|
| 1. Sämtliche Schlauchkupplungen an beiden Enden des Wärmeaustauschers lösen. Sämtliche Anschlüsse und die entsprechenden Rohre kennzeichnen. | 1 |
| 2. Die beiden hinteren Schrauben der Klemmbügel herausdrehen. | 2 |
| 3. Den Klemmbügel vorbiegen und den Wärmeaustauscher herausheben. | 3 |
| 4. Die Zentrummutter aus dem einen Ende des Wärmeaustauschers herausschrauben. | 4 |
| 5. Das gegossene Endstück mit einem Gummihammer aus dem Rohr schlagen. Die Lage des Endstückes im äusseren Rohr kennzeichnen. | 5 |
| 6. Die Rohre zum Reinigen herausziehen. Die Lage der Rohre gegeneinander anzeichnen. | |
| 7. Mit einer Bürste und saurem Reinigungsmittel reinigen. | |

Die Teile in umgekehrter Reihenfolge zusammenbauen. Wenn die Kennzeichnungen der Rohre durcheinandergekommen sind, sind die Rohre nach den Ziffern in der Zeichnung zu montieren.